THERMAL ANALYSIS OF REHEAT AND REGENERATIVE COMBINED CYCLE SYSTEM WITH CO-GENERATION

A project report submitted in partial fulfillment of the requirements for the award of the degree of

BACHELOR OF TECHNOLOGY

IN

MECHANICAL ENGINEERING

By

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CERTIFICATE

This is to certify that the project titled "THERMAL ANALYSIS OF REHEAT AND REGENERATIVE COMBINED CYCLE SYSTEM WITH CO-GENERATION" describes the bonafide work done by ***B.SAI SANDEEP, L.SHANKARA RAO, B.VINOD KUMAR, V.SANKAR NARAYANA*** *and* ***M.NAVEEN KUMAR****, in partial fulfilment for the award of B.TECH in Mechanical Engineering under the supervision and guidance of* ***M.S.S.SRINIVAS RAO*** *during the academic year 2018-2019.*

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INTERNAL EXAMINER: 15/4/19

15/4/19

EXTERNAL EXAMINER:

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ACKNOWLEDGEMENT

We are happy to present this report on "**Thermal Analysis of Reheat and Regenerative Combined Cycle System with Co-Generation**" in partial fulfilment of the requirement for the award of B.Tech., Degree in Mechanical Engineering.

We take this opportunity to express our deep and sincere indebtedness to our esteemed guide Mr.M.S.S.Srinivas Rao, Senior Assistant Professor, ANITS, a source of constant motivation in successfully completing our project.

We intend to express our thanks with sincere obedience to Prof.B.NagaRaju, Head of Mechanical engineering department and Prof.T.Subhramanyam, Principal for facilitating the execution of this work.

Lastly, we are grateful to one and all who have contributed either directly or indirectly in completion of this project.

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NOMENCLATURE

η	Efficiency
CV	Calorific Value (kJ/kg)
C_P	Specific Heat of Water (kJ/kgK)
ED	Exergy Destruction (kW)
EX	Exergy (kJ/kg)
E	Energy (kW)
E	Specific Energy (kJ/kg)
H	Specific Enthalpy (kJ/kg)
$\dot{m}$	Mass Flow Rate (kg/sec)
$\dot{m}_1$	Steam Bleed at 1st Extraction Point (kg/sec)
$\dot{m}_2$	Steam Bleed at 2nd Extraction Point (kg/sec)
P	Pressure (bar)
R	Characteristic Gas Constant (kJ/kgK)
S	Specific Entropy (kJ/kgK)
T	Temperature (°C)
W	Work Input or Output (kJ/kg)

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ABSTRACT

Simple cycle gas turbine engines have limited efficiencies, wherein efficiency can be enhanced by adopting different techniques like intercooling, reheating& regeneration. It is further observed that the performance can be boosted by extracting heat from the exhaust gases. This procedure when adopted has led to the development of combined cycle systems. The interest in combined cycles was aroused throughout the world in mid-1970's. During the last four decades a number of alternative combined cycle plants have been developed and are currently in operation.

A combined cycle can be driven using two working fluids or three working fluids. The topping cycle in either of the two cases is always a gas cycle. The bottoming cycle is generally steam driven. If three working fluids are used then the second cycle is steam driven and the bottoming cycle is driven by an organic or inorganic working fluid.

Improving the combined cycle efficiency has always been the primary objective of cycle analysis. In the present study a combined cycle system which utilizes two stage expansions with intermediate reheating in both gas and steam cycles is adopted. The system also includes two stage feed water heating. The bleeding of steam for the first feed water heater is from the H.P steam turbine and for the second feed water heater from the L.P steam turbine. The novelty of the system chosen lies in adopting co-generation. This is achieved by utilizing the waste heat of the gases exiting from heat recovery steam generator [HRSG].

The present analysis is dual prone i.e., based on both first and second laws. The second law analysis gives a clear picture of the irreversibilities associated with each process in the cycle. The parametric analysis consists in studying the effect of pressure ratio and maximum temperature of Brayton cycle on the overall cycle efficiencies, exergy destruction, power outputs, air-fuel ratio, specific fuel consumption and temperature of the water which is a by-product of co-generation. The energy and exergy efficiencies of the cycle are observed to be a strong function of the maximum temperature and pressure ratio. The exergy destruction of the cycle is found to decrease with pressure ratio. The effect of maximum temperature is to increase the total exergy destruction of the cycle. The total flow rate of gases in the cycle are observed to reach a maximum value at a pressure ratio of 3(approx) and thereafter decrease. The flow rate of

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steam is observed to be varying inversely with pressure ratio unlike the flow rate of gases. Further the steam flow rate is also directly dependent on turbine inlet temperature [TIT]. The analysis also includes evaluation of the ratio of power outputs of gas and steam cycles. This study shows that the gas and steam cycles produce equal power outputs at a pressure ratio of around 3 irrespective of the turbine inlet temperature. The power output of gas cycle is twice that of steam cycle at a pressure ratio of 8. The specific power output which is defined as the ratio of power output to fuel consumption rate varies with pressure ratio and also maximum temperature. A higher pressure ratio and maximum temperature seems to be beneficial. Since combustion is dependent on the percentage of the excess air available in the combustion chamber, it is necessary to evaluate the variation of excess air with pressure ratio and maximum temperature. The results are obvious by the fact that lower turbine inlet temperature has shown higher percentage of excess air. Further with an increase in pressure ratio, the amount of excess air diminishes. Co-generation is a part and parcel of the system. The water outlet temperature varies from 42°C to 45°C and the quantity of water varies from 40,000 lit/hr to 88,000 lit/hr which could be used for process application.

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CHAPTER-1

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1. INTRODUCTION

The combined cycle technology has attracted much attention by the researchers in the last few decades which adopts the Brayton cycle gas turbine and Rankine cycle steam turbine with air and water as respective working fluids to achieve efficient, reliable, and economic power generation. The exhaust gases from the Brayton cycle, hereafter termed as topping cycle are used as source of heat for steam generation in the steam cycle also called bottoming cycle.

The most commonly used means in industrial practice for generation of mechanical power is the utilization of gas and steam turbines. Different means have been employed by a lot of researchers to achieve better thermal efficiencies from the cycles. Combining two or more thermodynamic cycles result in improved overall efficiency reducing fuel costs. In stationary power plants, a widely used combination is a gas turbine operating by the Brayton cycle whose hot exhaust powers a steam power plant operating by the Rankine cycle. This is called a Combined Cycle power plant, and can achieve a thermal efficiency of around 60%. The Brayton cycle has a high source temperature and rejects heat at a temperature that is conveniently used as the energy source for the Rankine cycle plant.

Co-generative system may be defined as a unit which contains electricity production and regain of the thermo value of exhaust gases simultaneously. Efficient usage of energy becomes more important in our world where the fossil fuels are limited. Companies are paying attention to use energy efficiently by investing Co-generation plants where electricity and heat energy are used simultaneously. Co-generation system reduces the cost of energy in industries such as textile, folio, ceramic, chemistry, food and metal. Moreover, cogeneration applications on residential and industrial areas are giving promising results. Combined cycles augmented by co-generation could be considered as viable system which enhances the efficiency appreciably.

Generally, the performances of power plants are evaluated from first law perspective alone. But it is now a proven fact that first law analysis alone will not give a true picture of the irreversibilities associated with the cycle. This lacunae can be circumvented by applying second law analysis to the power plant systems. The second law analysis will not only quantify the irreversibilities in terms of exergy destruction but also stresses on the quality of energy. Hence power plants are generally evaluated from both first and second law perspectives.

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1.1 Energy:

Energy is the capacity to exert a force through a distance, and manifests itself in various forms. Engineering process involves the conversion of energy from one form to another, the transfer of energy from place to place, and storage in various forms, utilizing a working substance.

The unit of energy in the SI system is N-m (or) J (Joule). The energy per unit mass is the specific energy, the unit of which is J/kg.

$$Q = \Delta E + W$$

Where, ΔE is the increase in the energy of the system.

Common forms of energy include the kinetic energy of a moving system, the potential energy stored by system's position in a force field (gravitational, electric or magnetic), the elastic energy stored by stretching solid systems, the chemical energy released when a fuel burns, the radiant energy carried by light, and the thermal energy due to system's temperature.

1.2First Law of Thermodynamics:

The First Law of thermodynamics may be stated in several ways: The increase in internal energy of a closed system is equal to total of the energy added to the system. In particular, if the energy entering the system is supplied as heat and if energy leaves the system as work, the heat is accounted for as positive and the work as negative.

$$(\sum W)_{cycle} = J(\sum Q)_{cycle}$$

In the case of a thermodynamic cycle of a closed system, which returns to its original state, the heat Q_{in} supplied to the system in one stage of the cycle, minus the heat Q_{out} removed from it in another stage of the cycle, plus the work added to the system W_{in} equals the work that leaves the system W_{out}.

More specifically, the First Law encompasses several principles:

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The law of conservation of energy

This states that energy can be neither created nor destroyed. However, energy can change forms, and energy can flow from one place to another. A particular consequence of the law of conservation of energy is that the total energy of an isolated system does not change.

1.3Second Law of Thermodynamics:

The second law of thermodynamics indicates the irreversibility of natural processes, and, in many cases, the tendency of natural processes to lead towards spatial homogeneity of matter and energy, and especially of temperature. It can be formulated in a variety of interesting and important ways.

It implies the existence of a quantity called the entropy of a thermodynamic system. In terms of this quantity it implies that the second law is applicable to a wide variety of processes, reversible and irreversible. All natural processes are irreversible. Reversible processes are a useful and convenient theoretical fiction, but do not occur in nature.

A prime example of irreversibility is in the transfer of heat by conduction or radiation. It was known long before the discovery of the notion of entropy that when two bodies initially of different temperatures come into thermal connection, then heat always flows from the hotter body to the colder one.

The second law tells also about kinds of irreversibility other than heat transfer, for example those of friction and viscosity, and those of chemical reactions.

According to the second law of thermodynamics, in a theoretical and fictive reversible heat transfer, an element of heat transferred, δQ, is the product of the temperature (T), both of the system and of the sources or destination of the heat, with the increment (dS) of the system's conjugate variable, its entropy (S).

$$dQ = Tds$$

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1.4 First Law Efficiency (Energy Efficiency):

A common measure on energy is the first law efficiency. The first law efficiency is concerned only with the quantities of energy, disregards the forms in which the energy exists.

First law efficiency is defined as the ratio of energy output and energy input, while their difference is the energy loss. By reducing the energy loss the first law efficiency can be improved.

$$\eta_I = \frac{Energy\ output}{Energyinput}$$

1.5 Exergy:

The availability or exergy of a given system is defined as the maximum useful work that is obtainable in a process in which the system comes to equilibrium with its surroundings. Exergy is thus a composite property depending on the state of both the system and surroundings.

Whenever useful work is obtained during a process in which a finite system undergoes a change of state, the process must terminate when the pressure and temperature of the system have become equal to the pressure and temperature of the surroundings P_0 and T_0 i.e., when the system has reached the dead state. This represents the maximum useful work or exergy of the system.

$$W_{max} = [1 - T_0/T]Q = AE$$

For a given T_0, as T increases, the exergy increases. The exergy decreases as T decreases.

Thermodynamics is generally considered as subject of three E's- Energy, Entropy and Exergy. When a system undergoes a thermodynamic process, the energy is conserved. The entropy of the system and surroundings together increases due to irreversibilities and the exergy of the system decreases.

1.6 Exergy Analysis:

Exergy is a generic term for a group of concepts that defines the maximum possible work potential of a system. In an open flow system there are three types of energy transfers across the control surface namely work transfer, heat transfer and energy associated with mass transfer. The

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work transfer is equivalent to maximum work which can be obtained from that form of energy. The exergy of heat transfer (Q) from the control surface at temperature T is determined from maximum rate of conversion of thermal energy to work W_{max} is given by

$$W_{max} = Q\,(1-(T_0/T))$$

Exergy is defined as the maximum work that may be achieved by bringing a system into equilibrium with its environment. Exergy analysis is a method that uses the conservation of mass and conservation of energy principles together with the second law of thermodynamics for the analysis, design and improvement of energy systems. Exergy analysis is based on both first and second law of thermodynamics. Exergy analysis can clearly indicate the location of energy degradation in a process. Many engineers and scientists suggest that the thermodynamic performance of a process is best evaluated by performing an exergy analysis in place of conventional energy analysis because exergy analysis appears to provide more insights and to be more useful in furthering efficiency improvement efforts than energy analysis. The main purpose of exergy analysis is to identify the causes, types, location and to calculate magnitude of thermal losses.

1.7 Second Law Efficiency (Exergy Efficiency):

First law efficiency does not discriminate between the energies available at different temperatures. It is the second law of thermodynamics which provides a means of assigning a quality index to energy. The concept of available energy or exergy provides a useful measure of energy quality.

With this concept, it is possible to analyze means of minimizing the consumption of available energy to perform a given process, thereby ensuring the most efficient possible conversion of energy from the required task.

The second law efficiency of process is defined as the ratio of minimum available energy or exergy which must be consumed to do a task divided by the actual amount of energy consumed in performing the task.

$$\eta_{II} = \frac{minimum exergy intake to perform the given task}{actual exergy intake to perform the same task}$$

This definition is specifically used for devices like turbines, pumps, etc. which either produce or consume shaft work.

The second law efficiency can also be defined as the ratio of exergy recovered from a system to the exergy supplied. This definition holds for devices like heat exchangers, boilers, etc.

$$\eta_{II} = \frac{Exergyrecovered}{Exergysupplied}$$

1.7.1 Turbine:

The second law efficiencies for steady flow devices like turbines, compressors, heat exchangers and mixing chambers can be determined from their definitions. For a work producing devices such as turbine, the second law efficiency is defined as

$$\eta_{IIturbine} = \left(\frac{W_u}{W_{rev}}\right) = \left(\frac{W_{act}}{W_{max}}\right)$$

Where W_u is the actual useful work and W_{rev} is the reversible work

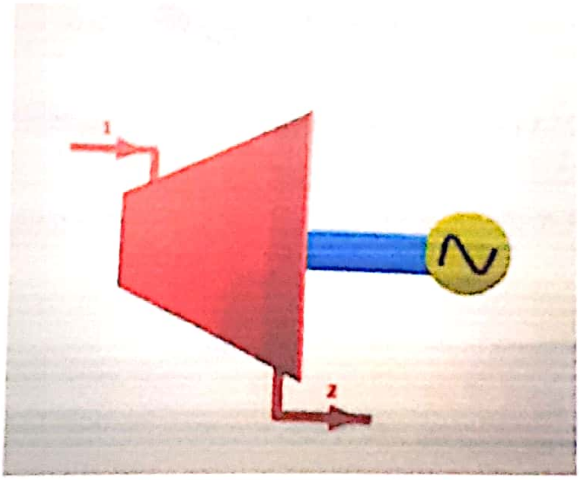

Fig 1.1 Block Diagram of Turbine

The actual work of the turbine

$$W_{act} = h_1 - h_2$$

The reversible work can be determined by setting the exergy destructions to zero in the exergy balance relation. For one inlet- one exit steady flow devices, such as turbine, it is

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$$W_{max} = (h_1 - h_2) - T_0(S_1 - S_2)$$

1.7.2 Pump:

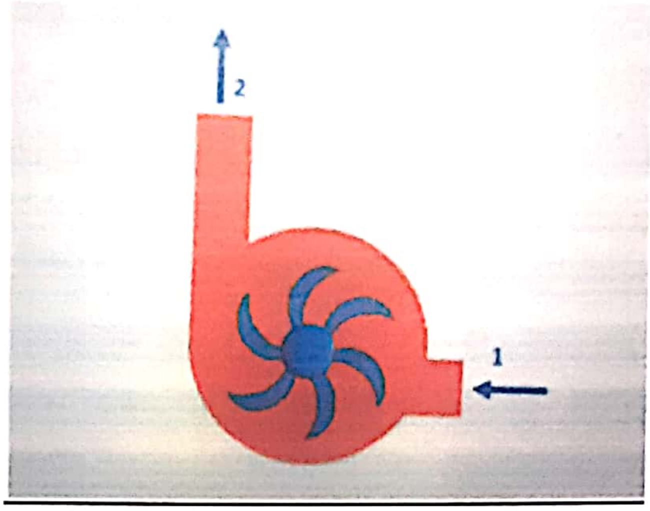

Fig 1.2 Block Diagram of Pump

$$W_{act} = h_2 - h_1$$

$$W_{min} = (h_2 - h_1) - T_0(S_2 - S_1)$$

The second law efficiency of a pump can e determined using a similar method. It is given as

$$\eta_{IIpump} = \frac{W_{min}}{W_{act}}$$

1.7.3 Heat Exchanger:

Since no work is involved in heat exchangers and mixing chambers, their second law efficiencies are defined as the ratio of exergy recovered to exergy supplied. A heat exchanger with two unmixed fluid streams is shown. Hot stream at state-1 and state-2, cold stream enters at state 3 and leaves at state-4.

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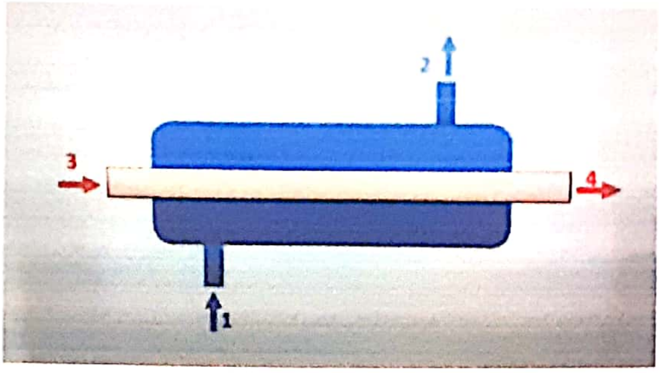

Fig 1.3 Block Diagram of Heat Exchanger

During the heat exchange process, exergy supplied equals the exergy lost in the hot stream and exergy recovered equals the exergy gained in the cold stream. Thus, the second law efficiency of a heat exchanger is

$$\eta_{\mathrm{II\,he}} = \frac{Exergy increase of cold fluid}{Exergy decrease of hot fluid}$$

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2. BRAYTON-RANKINE COMBINED CYCLE

2.1 Brayton Cycle:

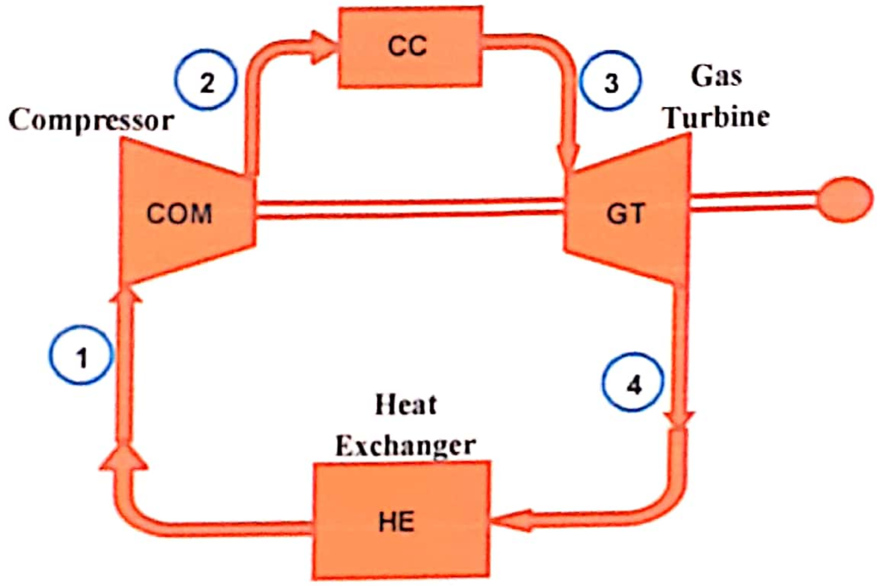

Fig 2.3 Layout of Brayton Cycle

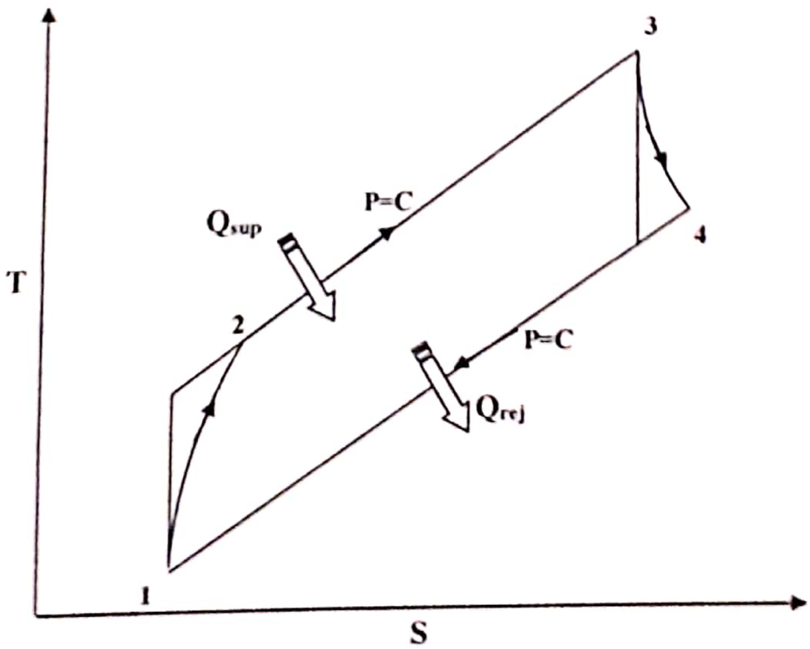

Fig 2.4 T-S plot of Brayton Cycle

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The two major application areas of gas turbine engines are aircraft propulsion and electric power generation. This section introduces the ideal cycle for gas turbine engine - Brayton cycle. Brayton cycle consists the following four processes.

1-2 Isentropic Compression:

A compressor takes in fresh ambient air, compresses it to a higher temperature and pressure.

$$W_{com} = h_2 - h_1$$

2-3 Constant Pressure Heat Addition:

Fuel and the higher pressure air from compressor are sent to a combustion chamber, where fuel is burned at constant pressure. The resulting high temperature gases are sent to a turbine.

$$q_{in} = h_3 - h_2$$

3-4 Isentropic Expansion:

The high temperature gases expand to the ambient pressure in the turbine and produce power.

$$W_{turbine} = h_3 - h_4$$

4-1 Constant Pressure Heat Rejection:

Heat is rejected at a constant pressure process to the surroundings if it is an open cycle. A heat exchanger is used if it is a closed cycle which rejects heat.

$$q_{out} = h_4 - h_1$$

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2.2Rankine cycle:

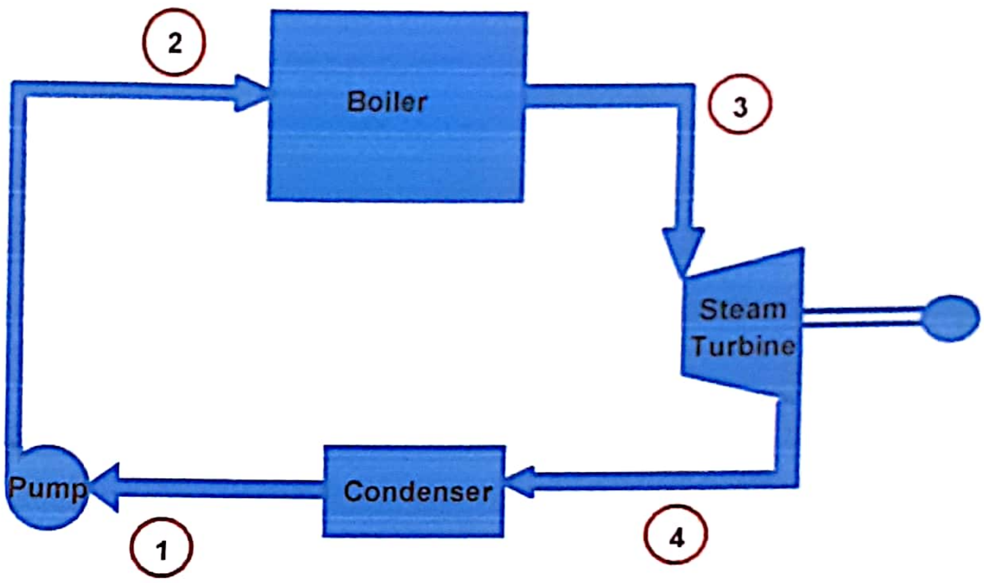

Fig 2.3 Layout of Rankine Cycle

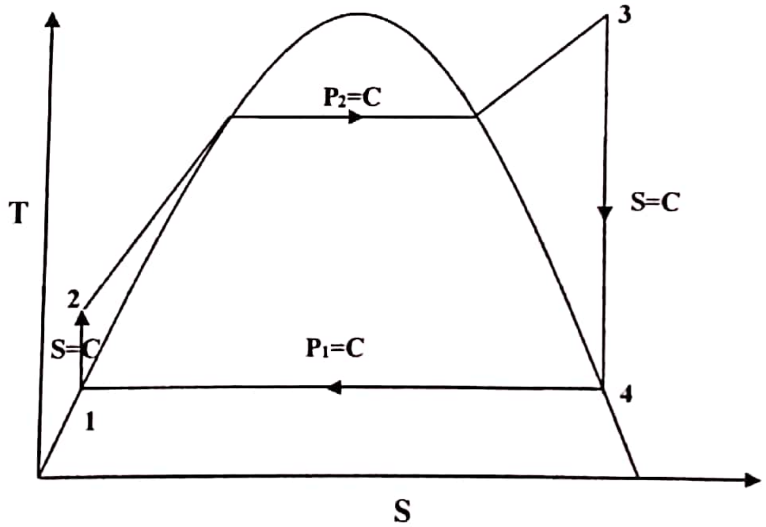

Fig 2.4 T-S plot of Rankine Cycle

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The Rankine cycle is an ideal cycle if water passes through the four components without irreversibilities and pressure drops. The ideal Rankine cycle consists of the following four processes.

1-2: Isentropic Compression:

The working fluid is pumped from low pressure to high pressure. As the fluid is a liquid at this stage, the pump requires little input energy. The energy balance in the pump is

$$W_{pump} = h_2 - h_1$$

2-3: Constant Pressure Heat Addition:

Liquid water enters the boiler and is heated to superheated state in the boiler. The energy balance in the boiler is

$$q_{in} = h_3 - h_2$$

3-4: Isentropic Expansion:

Steam from the boiler which has an elevated temperature and pressure expands through the turbine to produce work and then is discharged to the condenser with relatively low pressure. Neglecting heat transfer with the surroundings, the energy balance in the turbine is

$$W_{turbine} = h_3 - h_4$$

4-1: Constant Pressure Heat Rejection:

Steam from the turbine is condensed to liquid water in the condenser. The energy balance in the condenser is

$$q_{out} = h_4 - h_1$$

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.3 Brayton-Rankine Combined Cycle with Reheating, Re-Generation and Co-Generation:

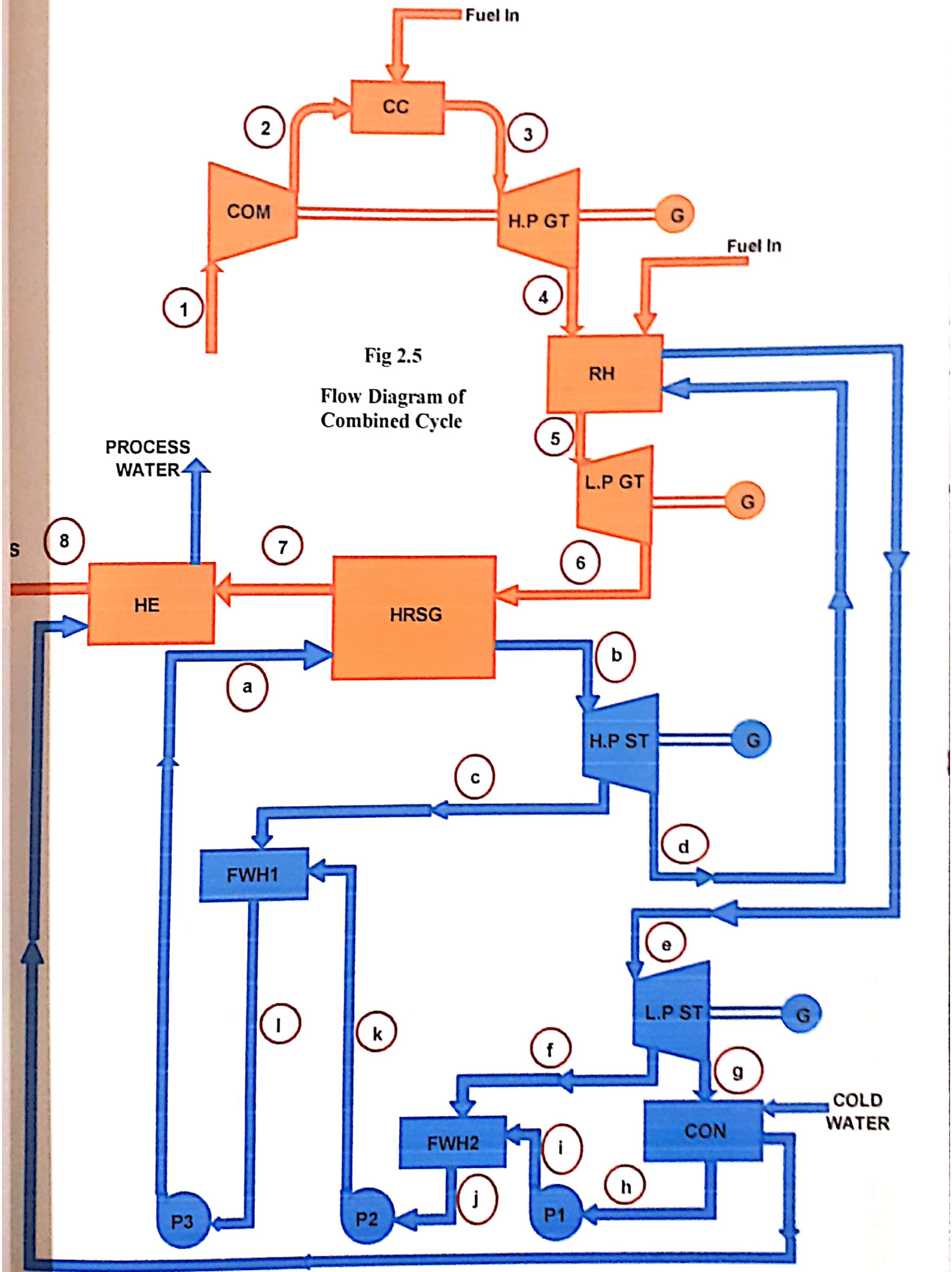

Fig 2.5

Flow Diagram of Combined Cycle

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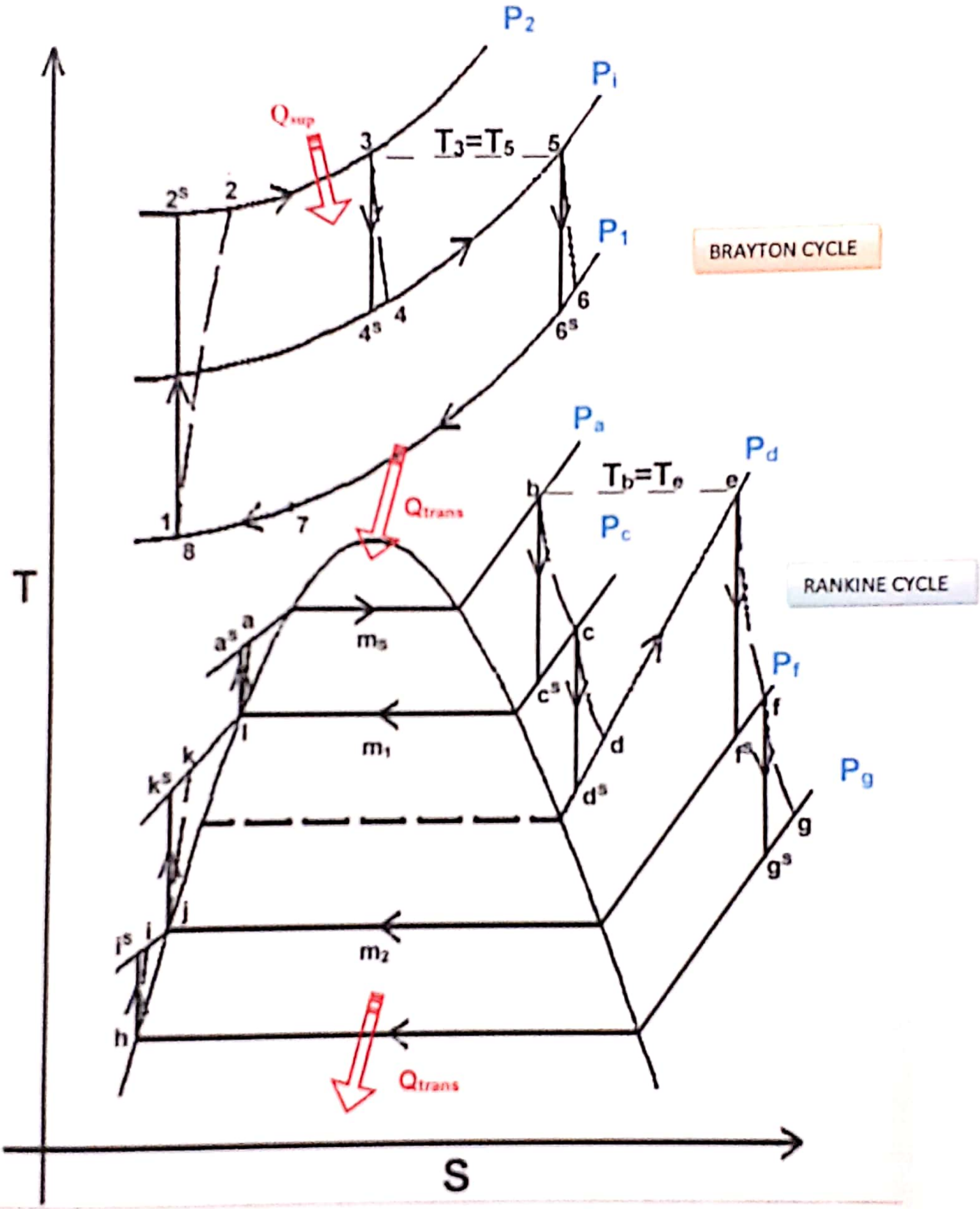

Fig 2.6 T-S PLOT OF COMBINED CYCLE

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2.4 Significance of Pinch Point:

It refers to the states of the liquid and gas when their temperature difference is minimum. The temperature difference corresponding to this point is known as pinch point temperature difference. This point has special significance as it is necessary to consider this in the heat balance equations. Generally for thermal design, when one of the fluid is undergoing a phase change, the pinch point temperature difference is assumed. This is as indicated in the figure below

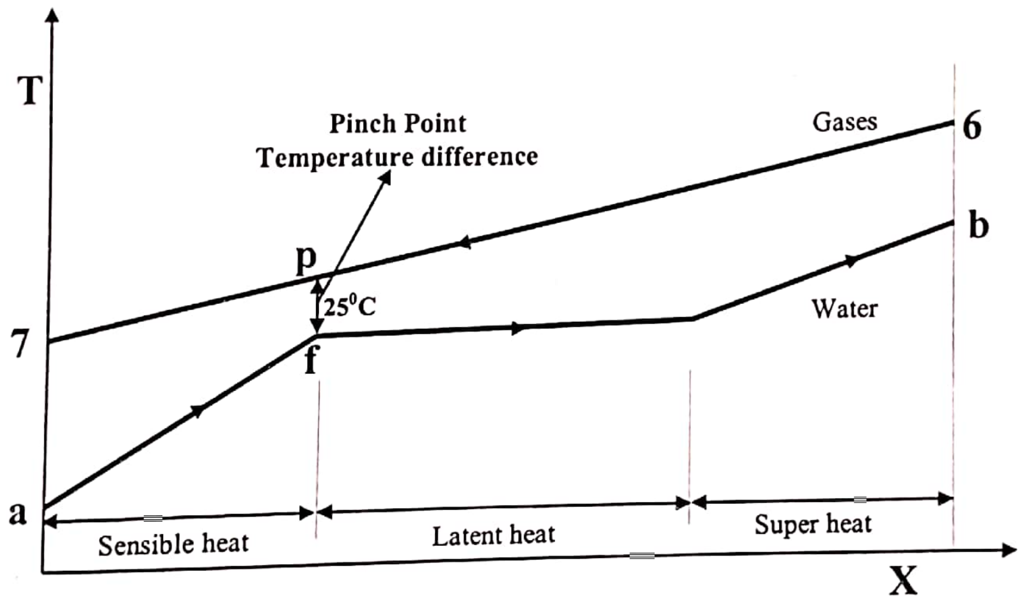

Fig 2.7 Temperature profiles of hot gases and water

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3. LITERATURE REVIEW

Combined cycle power plants are extensively in use, as they offer higher efficiencies then simple vapour cycles. The essential topography of combined cycles is the waste heat from the gas cycle being utilized to drive the vapour cycle. This has a positive impact on the performance of the power plant. Such cycles have been subjected to intensive research as the literature review indicates. A few of the works done so far are illustrated hereunder:

Nikhil Dev et.al [1] reviewed the various types of combined cycles including repowering, integrated gasification and other advance systems. They concluded that high thermal efficiency, low installed cost, low operation and maintenance costs, operating flexibility and high reliability are the features of combined cycle systems.

A.K.Tiwari et.al [2] reviewed the effect of various operating parameters on the performance of the combined cycle power plant. Based on the review, the authors concluded that the major operating parameters that affect the performance are the turbine inlet temperature, compressor pressure ratio, pinch point temperature, ambient temperature and the number of stages of expansion.

Padma DharGarg et.al [3] analyzed a combined cycle power plant with co-generation. They investigated the effect of gas turbine pressure ratio, gas turbine inlet temperature, process heat load and ambient conditions. It is demonstrated that a combined cycle co-generation unit operates more efficiently and produces less carbon dioxide than too separate, power production and process heat systems. They reported that the exergy loss is significantly affected by the pressure ratio and turbine inlet temperature.

DeryaBurcuOzkan et.al [4] reported exergy analysis of an acting co-generation plant. The authors study was aimed to determine the influence of environmental conditions on the second law efficiency of plant.

Nikhil Dev& Rajesh Attri [5] made a comparative study of different combined power plant streams. Based on the study the authors suggested that co-generation power plant systems [CPPS] are capable of meeting the power shortages during peak and off peak hours.

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Koushiket.al [6] reviewed in detail the different studies on the thermal power plants over the years. This has thrown light on the scope of further research. The author's suggestions for improvement in first and second law efficiencies are worth noting.

Habibet.al [7] set out a procedure for optimization of reheat pressure level in reheat, regenerative thermal power plants based on both first & second laws. They also performed optimization studies on the 2nd law efficiency of steam generator, turbine cycle and the overall plant.

Olikeret.al [8]. The design features & performance characteristics of co-generative turbine limits for combined electric generation and district heat supply are presented. The main thrust of the study was to relate the heat load duration curve for urban heat supply to the different modes of operation of the turbine.

Gogoi et.al [9]. The work presented by the author is related to combined reheat, regenerative steam turbine based power cycles combined with vapour absorption refrigeration system. A detailed exergy analysis of the system is presented and the effect of fuel flow rate, boiler pressure, cooling capacity on both component and total system irreversibility is analyzed.

Mohanty et.al [10] analyzed the performance of a combined cycle gas turbine under varying operating conditions. The effect of various operating parameters such as the maximum pressure and temperature of Rankine cycle etc., on the performance is investigated.

Martina Rauch et.al [11] reported the optimization of combined Brayton-Rankine cycle with respect to the total thermal efficiency. The effect of pitch point temperature difference, dryness fraction at the exit of steam turbine and other important parameters was mathematically simulated.

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The literature review accomplished herein reflected the fact that there is still a wide scope for research in the area of combined cycle power plants. Very few authors have studied combined cycle power plants integrated with co-generation. Moreover the works testified in the literature ascertained the gaps with reference to the complete evaluation of combined cycle configurations.

It is therefore decided by the authors of this work that a detailed thermal analysis of combined cycle systems integrated with reheating, regeneration and co-generation could be an inspiring topic for study.

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4. THERMODYNAMIC ANALYSIS

Statement of the Problem:

Thermal analysis of Reheat and Regenerative combined cycle power system integrated with co-generation is the objective of the work. As a part of this objective the effect of pressure ratio and turbine inlet temperature of the topping cycle on the performance parameters like first and second law efficiencies, exergy destruction, specific fuel consumption, air-fuel ratio etc., are studied.

Assumptions:

- The working fluid of the topping cycle is a perfect gas and its properties are invariant.
- Pressure drop due to viscous effects, turbulence and variations in cross-sections of the ducting are negligible.
- The combustion phenomenon is assumed to be isobaric process.
- All the thermodynamic process are considered as irreversible.
- The efficiencies of the main devices are assumed as η_{com}=90%,η_{cc}=90%, η_{gt}=90%,η_{st}=90%,η_{hrsg}=90%,η_{rh}=90%,η_{con}=90%, η_{p}=85%,η_{he}=90%.
- The bleeding of steam for re-generation is at constant pressure.
- The changes of kinetic energy and potential energy of the working fluids are neglected.
- The combustion process is carried out with excess air.
- The fuel used is methane with calorific value of 50,010 kJ/kg and specific exergy of51,888 kJ/kg.
- The pinch point temperature difference is assumed as 25°C.
- The dead state temperature is assumed as 30°C.

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4.1 Combustion Equation:

The fuel considered for the analysis is METHANE.

$$CH_4+2O_2 \longrightarrow CO_2+2H_2O$$

1 mole of CH_4 requires $\longrightarrow$ 2 moles of O_2

16 kg of CH_4 $\longrightarrow$ 64 kg of O_2

1 kg of CH_4 $\longrightarrow$ 4 kg of O_2

1 kg of air contains $\longrightarrow$ 23.3% O_2 by weight

$\longrightarrow$ 0.233 kg of O_2

Amount of fuel that can be burnt with 1 kg of air = 0.233/4

= 0.05825 kg

In the thermodynamic analysis, it is pertinent to cross check the amount of fuel calculated with the above obtained value i.e., 0.05825 kg. The amount of fuel obtained through calculations should be less than the maximum value calculated above.

4.2 Determination of Chemical Exergy of Methane (CH_4):

$$CH_4+2O_2 \longrightarrow CO_2+2H_2O$$

The chemical exergy of the fuel is evaluated from the basic formula

Specific Exergy $e_f=[\bar{g}_{(CH4)}+2\ \bar{g}_{(O2)}-\bar{g}_{(CO2)}-2\ \bar{g}_{(H2O)}]+\bar{R}T_0\ \ln[(x_{(O2)})^2/\{(x_{(O2)})^1\ (x_{(O2)})^2\}]$

Where, $\bar{g}$-the chemical exergy of the constituent,

$\bar{R}$-Universal Gas Constant,

T_0-Dead State Temperature,

x -Mole fraction of the constituent in the environment air.

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The composition of the environment on molar basis is N_2-75.67%, O_2-20.35%, H_2O-3.12%, CO_2-0.03% and others-0.83%.

Specific Exergy = $-50751+2(0)-1(-394380)-2(-228590)+8.314(29815)\ \ln[(0.2035)^2/\{(0.0003)^1 (0.0312)^2\}]$

$= 830214.368$ kJ/kg mol

Specific Exergy of Methane= 830214.368/16 = 51888 kJ/kg

4.3 ENERGY ANALYSIS:

4.3.1 BRAYTON CYCLE (Topping Cycle):

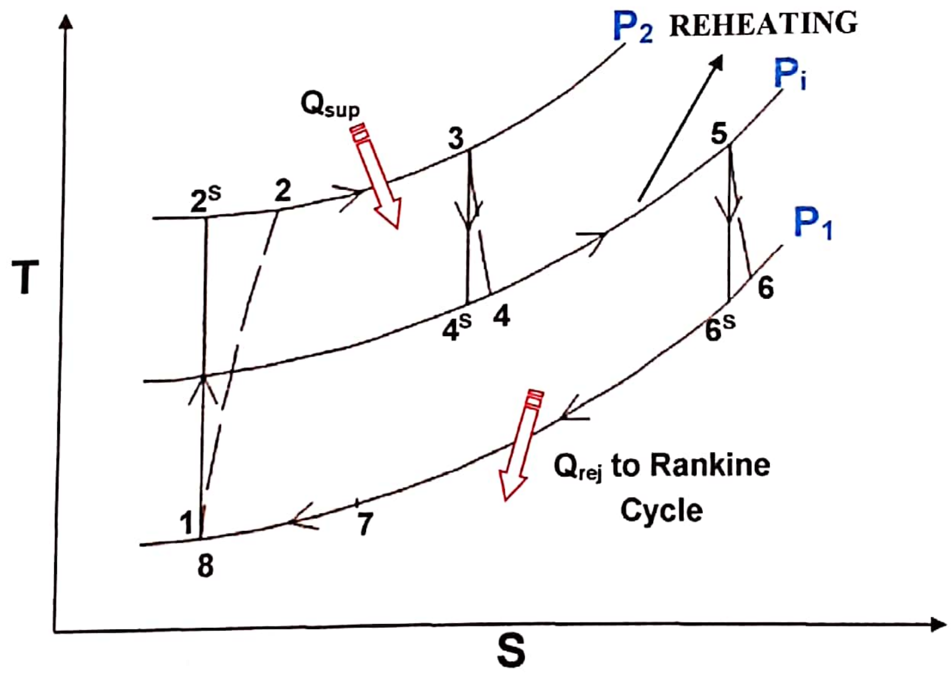

Fig 4.1 T-S plot of Brayton Cycle with Reheating

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1) Compressor(Process 1-2):

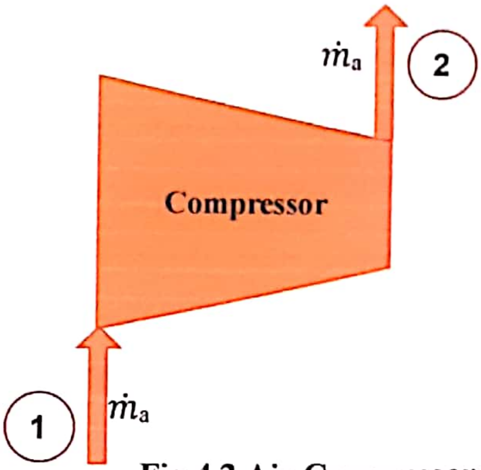

Fig 4.2 Air Compressor

Considering Isentropic process **1-2^s**

$$(T_{2S}/T_1) = (P_2/P_1)^{\varphi} = (r_p)^{\varphi}$$

Where,

$$\varphi=\gamma -1/\gamma$$

$$\Psi=(r_p-1)/\eta_c$$

$$W_c= h_2-h_1 =\Psi c_p T_1$$

2) Combustion Chamber(Process 2-3):

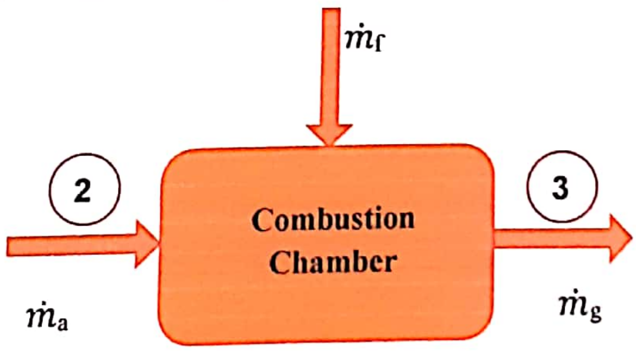

Fig 4.3 Combustion Chamber

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Energy balance gives:

$$\dot{m}_a h_2 + \dot{m}_f CV(\eta_{cc}) = \dot{m}_g h_3$$

$$\dot{m}_f = T_3 - T_1(1+\Psi)/\eta_{cc}\rho - T_3$$

Where,

$$\rho = CV/C_p$$

3) H.P Gas Turbine(Process 3-4):

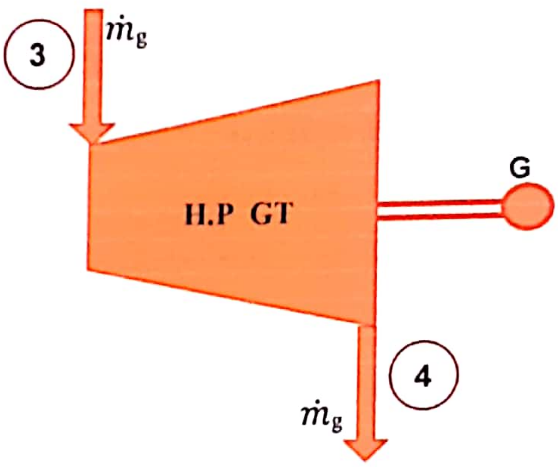

Fig 4.4 H.P Gas Turbine

Considering Isentropic process 3-4'

$$(T_3/T_{4S}) = (P_2/P_i)^{\varphi} = (r_{p1})^{\varphi}$$

$$\varphi = \gamma - 1/\gamma$$

The optimum pressure ratio for maximum work output is

$$P_1/P_i = P_i/P_2$$

$$P_i = (P_1/P_2)^{\wedge 0.5}$$

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$\eta_{gt1}=(h_3-h_4)/(h_3-h_{4S})$

$W_{gt1} = \eta_T C_P T_3[1-(1/r_{p1})^{\varphi}]$

4) Combined Reheater(Processes 4-5, d-e):

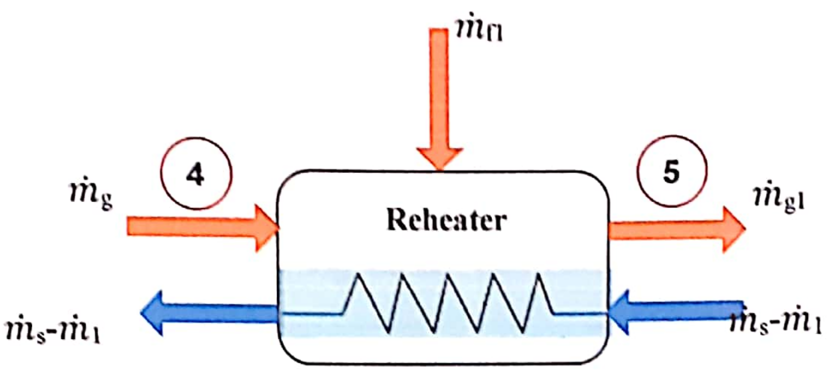

Fig 4.5 Combined Reheater

Energy balance gives:

$$(\dot{m}_{g1}h_5 - \dot{m}_g h_4)+\dot{m}_s(h_e-h_d)=\dot{m}_f CV(\eta_{cc})$$

Simplifying the expression, the mass flow rate of fuel in the re-heater is

$$\dot{m}_{f1}= (1+\dot{m}_f)\{[(T_5-T_4)+\beta]/ [(\eta_{cc}\rho -T_5)+\beta]\}$$

Where,

$$\beta= (T_6-T_P) [(h_e-h_d)/ (h_b-h_f)]$$

$$T_6=T_4=T_3(1-\alpha)$$

$$\alpha= \eta_{gt1}[1-(1/(1/r_{p1})^{\varphi})]$$

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5) L.P Gas Turbine(Process 5-6):

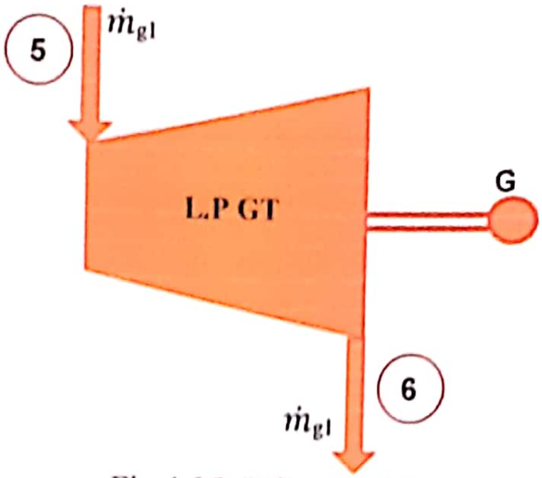

Fig 4.6 L.P Gas Turbine

Considering Isentropic process 5-6'

$$\eta_{gt2}=(h_5-h_6)/(h_5-h_{6S})$$

$$W_{gt2}=\eta_T C_P T_3[1-(1/r_{p1})^{\varphi}]$$

4.4.2 RANKINE CYCLE:

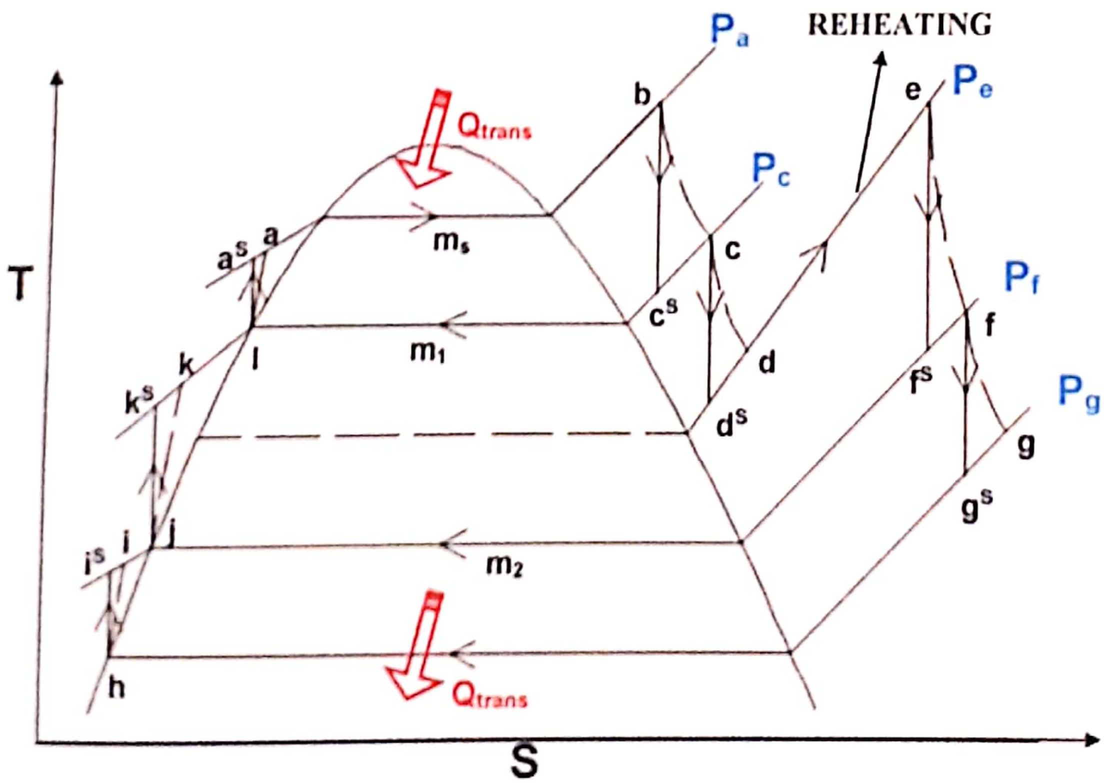

Fig 4.7. T-S plot of Rankine Cycle with Reheating and Regeneration

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1) Heat Recovery Steam Generator(Processes 6-7, a-b):

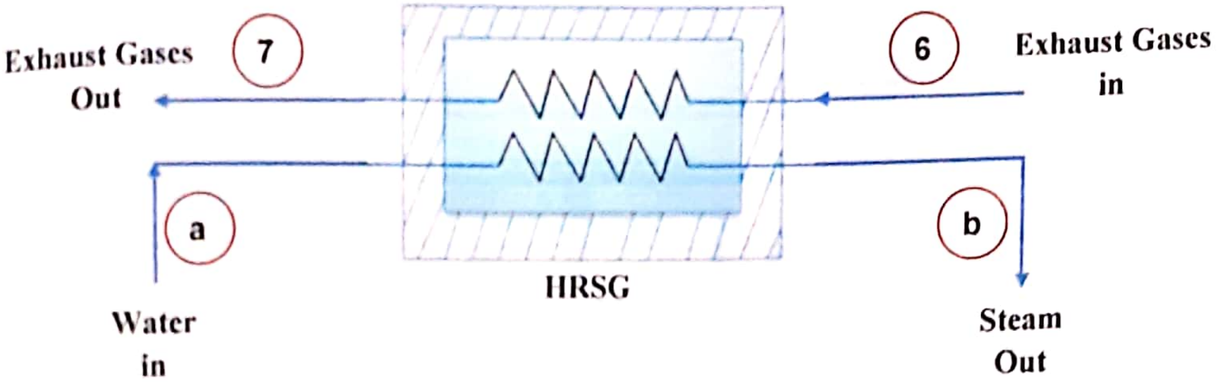

Fig 4.8 HRSG

Pinch Point Temperature Difference is assumed as 25°C

$$T_P = T_{sat} + 25$$

Energy balance in the latent and superheat region gives:

$$\eta_{hrsg}\dot{m}_g(h_6 - h_p) = \dot{m}_s(h_b - h_{f1})$$

Simplifying the expression, the mass flow rate of steam is

$$\dot{m}_s = \dot{m}_g C_p(T_6 - T_p)\,\eta_{hrsg} / (h_b - h_{f1})$$

Energy balance in the sensible heat region gives:

$$\eta_{hrsg}\dot{m}_{g1}(h_p - h_7) = \dot{m}_s(h_{f1} - h_a)$$

Simplifying the expression,

$$T_7 = T_P - [\dot{m}_s(h_{f1} - h_a)/(\dot{m}_g C_p \eta_{hrsg})]$$

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2) H.P Steam Turbine(Process b-c):

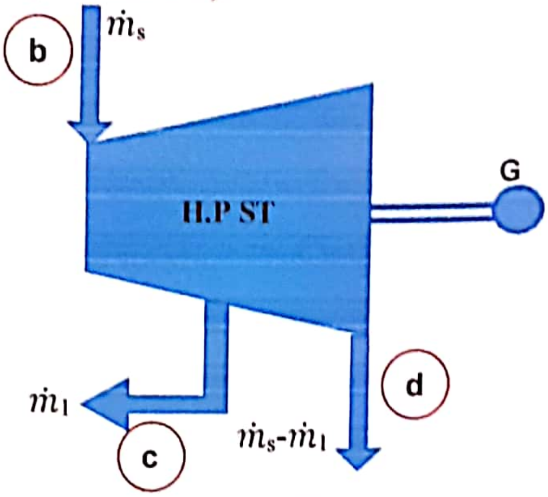

Fig 4.9 H.P Steam Turbine

Considering Isentropic process **b-c' & c-d'**

$$\eta_{st1} = (h_b - h_c)/(h_b - h_{cs})$$

$$\eta_{st1} = (h_c - h_d)/(h_c - h_{ds})$$

$$W_{st1} = \dot{m}_s(h_b - h_c) + (\dot{m}_s - \dot{m}_1)(h_c - h_d)$$

3) L.P Steam Turbine(Process e-f):

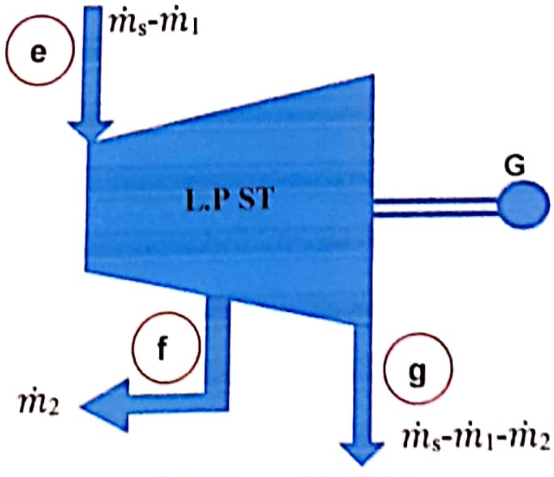

Fig 4.10 L.P Steam Turbine

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Considering Isentropic process e-f' & f-g'

$$\eta_{st2} = (h_e - h_f)/(h_e - h_{fs})$$

$$\eta_{st2} = (h_f - h_g)/(h_f - h_{gs})$$

$$W_{st2} = (\dot{m}_s - \dot{m}_1)(h_e - h_f) + (\dot{m}_s - \dot{m}_1 - \dot{m}_2)(h_f - h_g)$$

4) Feed Water Heater 1(Process c-l, k-l):

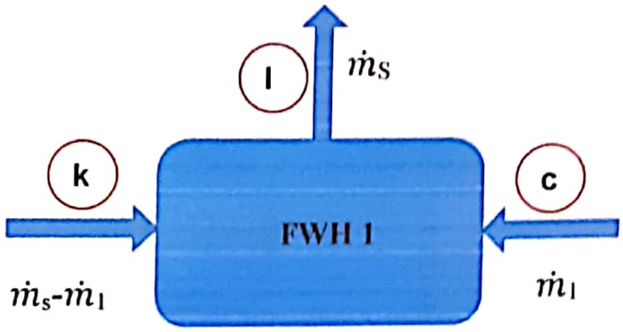

Fig 4.11 FWH1

Energy balance gives:

$$\dot{m}_1 h_c + (\dot{m}_s - \dot{m}_1) h_k = \dot{m}_s h_l$$

Simplifying the expression

$$\dot{m}_1 = \dot{m}_s (h_l - h_k)/h_c - h_k$$

5) Feed Water Heater 2(Process f-j, i-j):

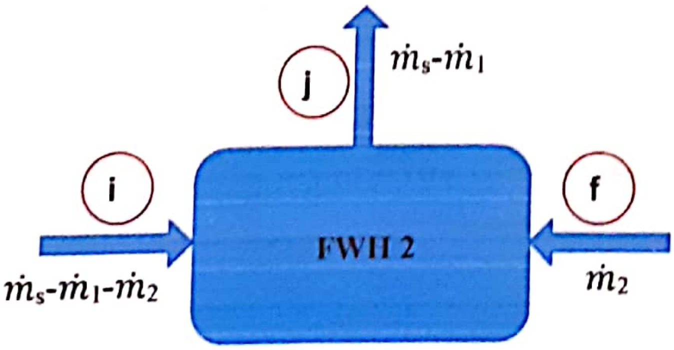

Fig 4.12 FWH2

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Energy balance gives:

$$\dot{m}_2 h_f + (\dot{m}_s - \dot{m}_1 - \dot{m}_2) h_i = (\dot{m}_s - \dot{m}_1) h_j$$

Simplifying the expression

$$\dot{m}_2 = (\dot{m}_s - \dot{m}_1)\,(h_j - h_i)/(h_f - h_i)$$

6) Condenser(Process g-h):

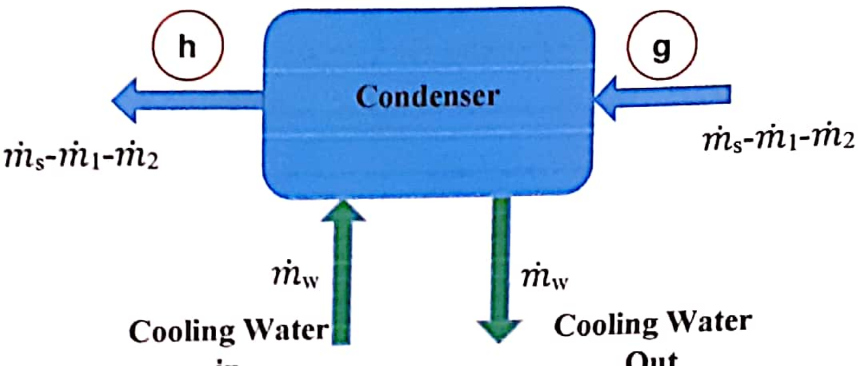

Fig 4.13 Condenser

Energy balance gives:

$$\eta_{con}(\dot{m}_s - \dot{m}_1 - \dot{m}_2)(h_g - h_h) = \dot{m}_w C_{pw}(T_{w2} - T_{w1})$$

The mass flow rate of cooling water is

$$\dot{m}_W = \eta_c(\dot{m}_s - \dot{m}_1 - \dot{m}_2)(h_g - h_h)/(t_1 - t_2)$$

7) Feed Water Pump 1(Process h-i):

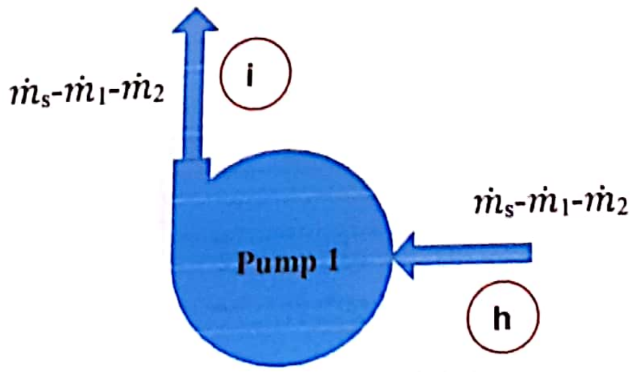

Fig 4.14 Feed Water Pump 1

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Isentropic Pump Work

$$W_{P1} = 100\vartheta_h(P_f\text{-}P_g)$$

Actual Pump Work

$$W_{P1(act)} = W_{P1}/\,\eta_p$$

8) Feed Water Pump 2(Process j-k):

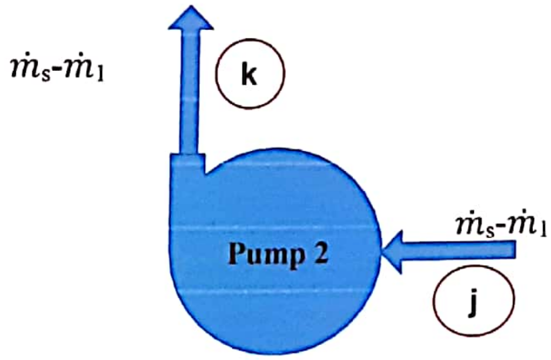

Fig 4.15 Feed Water Pump 2

Isentropic Pump Work

$$W_{P2} = 100\vartheta_j(p_c\text{-}p_f)$$

Actual Pump Work

$$W_{P1(act)} = W_{P1}/\,\eta_p$$

9) Feed Water Pump 3(Process l-a):

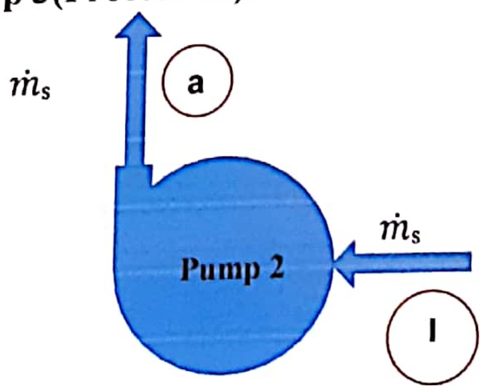

Fig 4.16 Feed Water Pump3

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Isentropic Pump Work

$$W_{P3} = 100\vartheta_l(P_a\text{-}P_c)$$

Actual Pump Work

$$W_{P1(act)} = W_{P1} / \eta_p$$

10) **Co-Generative heat exchanger(Process 7-8):**

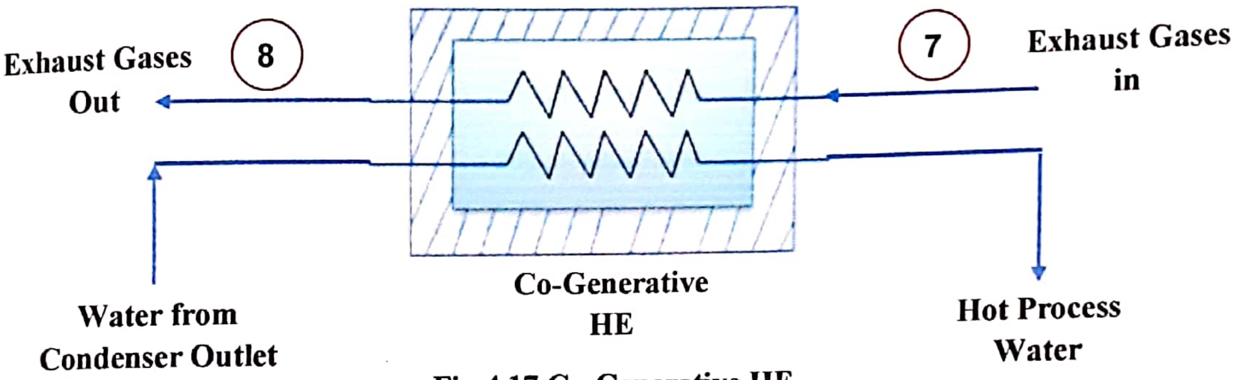

Fig 4.17 Co-Generative HE

Energy balance gives:

$$\eta_{he}\dot{m}_{gl}(h_g\text{-}h_h)=\dot{m}_w(h_{process}\text{-}h_{tw2})$$

$$h_{process}=[\ \eta_{he}\dot{m}_{gl}(h_g\text{-}h_h)/\dot{m}_w]+h_{tw2}$$

4.4 EXERGY ANALYSIS:

4.4.1 BRAYTON CYCLE:

1) **Compressor(Process 1-2):**

Considering Isentropic process **1-2^s**

$$W_{actcom}=h_2\text{-}h_1$$

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$$W_{mincom}=(h_2-h_1)-T_0(s_2-s_1)$$

Where,

$$s_2-s_1=C_p\ln(T_2/T_1)-R\ln(P_2/P_1)$$

$$ED_{com}=W_{actcom}-W_{mincom}$$

$$\eta_{exergycom}=W_{mincom}/W_{actcom}$$

2) Combustion Chamber(Process 2-3):

$$E_{supcc}=\dot{m}_f(e_f)$$

$$E_{reccc}=(1+\dot{m}_f)h_3-h_2-[T_0(1+\dot{m}_f)s_3-s_2]$$

Where,

$$s_3=C_p\ln(T_3/T_0)-R\ln(P_2/P_1)$$

$$s_2=C_p\ln(T_2/T_0)-R\ln(P_2/P_1)$$

$$ED_{cc}=E_{supcc}-E_{reccc}$$

$$\eta_{exergycc}=E_{reccc}/E_{supcc}$$

3) H.P Gas Turbine (Process 3-4):

Considering Isentropic process 3-4'

$$W_{actgt1}=h_3-h_4$$

$$W_{maxgt1}=(h_3-h_4)-T_0(s_3-s_4)$$

Where,

$$s_3-s_4=C_p\ln(T_3/T_4)-R\ln(P_3/P_4)$$

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$$ED_{gt1} = W_{maxgt1} - W_{actgt1}$$

$$\eta_{exergygt1} = W_{actgt1} / W_{maxgt1}$$

4) Reheater(Processes 4-5, d-e):

$$E_{suprh} = \dot{m}_{f1}(e_f) + \dot{m}_g(e_4) + \dot{m}_s(e_d)$$

$$E_{recrh} = \dot{m}_{g1}(e_5) + \dot{m}_g(e_e)$$

Where,

$$e_4 = h_4 - T_0 S_4$$

$$S_4 = C_p \ln(T_4/T_0) - R\ln(P_4/P_0)$$

$$e_d = h_d - T_0 S_d$$

$$ED_{rh} = E_{suprh} - E_{recrh}$$

$$\eta_{exergyrh} = E_{recrh} / E_{suprh}$$

5) L.P Gas Turbine(Process 5-6):

Considering Isentropic process 5-6'

$$W_{actgt2} = h_5 - h_6$$

$$W_{maxgt2} = (h_5 - h_6) - T_0(s_5 - s_6)$$

Where,

$$s_5 - s_6 = C_p \ln(T_5/T_6) - R\ln(P_5/P_6)$$

$$ED_{gt2} = W_{maxgt2} - W_{actgt2}$$

$$\eta_{exergyT2} = W_{actgt2} / W_{maxgt2}$$

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4.3.2 RANKINE CYCLE:

1) Heat Recovery Steam Generator(Processes 6-7 and a-b):

$$E_{suphrsg}=\dot{m}_{g1}[C_p(T_6-T_7)- T_0(s_6-s_7)]$$

$$E_{rechrsg}=\dot{m}_S(e_b-e_a)$$

Where,

$$s_6-s_7= C_p\ln(T_6/T_7)$$

$$e_b=h_b- T_0S_b$$

$$e_a=h_a- T_0S_a$$

$$ED_{hrsg}= E_{suphrsg}- E_{rechrsg}$$

$$\eta_{exergyhrsg}= E_{rechrsg}/ E_{suphrsg}$$

2) H.P Steam Turbine(Process b-d):

$$W_{actst1}=\dot{m}_s(h_b-h_c)+(\dot{m}_s-\dot{m}_1)(h_c-h_d)$$

$$W_{maxst1}=\dot{m}_s(e_b-e_c)+(\dot{m}_s-\dot{m}_1)(e_c-e_d)$$

Where,

$$e_c=h_d- T_0S_c$$

$$ED_{st1}= W_{maxst1}-W_{actst1}$$

$$\eta_{exergyst1}= W_{actst1}/W_{maxst1}$$

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3) L.P Steam Turbine(Process e-f):

$$W_{actst2}=(\dot{m}_s-\dot{m}_1)(h_e-h_f)+(\dot{m}_s-\dot{m}_1-\dot{m}_2)(h_f-h_g)$$

$$W_{maxst1}=(\dot{m}_s-\dot{m}_1)(e_e-e_f)+(\dot{m}_s-\dot{m}_1-\dot{m}_2)(e_f-e_g)$$

Where,

$$e_e=h_e-T_0S_e$$

$$e_f=h_f-T_0S_f$$

$$e_g=h_g-T_0S_g$$

$$ED_{st2}=W_{maxst2}-W_{actst2}$$

$$\eta_{exergyst2}=W_{actst2}/W_{maxst2}$$

4) Feed Water Heater 1(Processes c-l, k-l):

$$E_{supfwh1}=(\dot{m}_s-\dot{m}_1)e_k+\dot{m}_1e_c$$

$$E_{recfwh1}=\dot{m}_se_l$$

Where,

$$e_k=h_k-T_0S_k$$

$$e_l=h_l-T_0S_l$$

$$ED_{fwh1}=E_{supfwh1}-E_{recfwh1}$$

$$\eta_{exergyfwh1}=E_{recfwh1}/E_{supfwh1}$$

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5) **Feed Water Heater 2(Process f-j, i-j):**

$$E_{supfwh2}=(\dot{m}_s-\dot{m}_1-\dot{m}_2)e_i+\dot{m}_2e_f$$

$$E_{recfwh2}=(\dot{m}_s-\dot{m}_1)e_j$$

Where,

$$e_i=h_i- T_0S_i$$

$$e_f=h_f- T_0S_f$$

$$ED_{fwh2}=E_{supfwh2}- E_{recfwh2}$$

$$\eta_{exergyfwh2}= E_{recfwh2}/E_{supfwh2}$$

6) **Condenser(Process g-h):**

$$E_{supcon}=(\dot{m}_s-\dot{m}_1-\dot{m}_2)(e_g-e_h)$$

$$E_{reccon}=\dot{m}_w(e_{w2}-e_{w1})$$

Where,

$$s_{w2}-s_{w1}= C_{pg}\ln(T_{w2}/T_{w1})- R\ln(P_2/P_1)$$

$$e_{w2}-e_{w1}= C_{pg}T_{w2}- C_{pg}T_{w1}-T_0(s_{w2}-s_{w1})$$

$$ED_{con}=E_{supcon}- E_{reccon}$$

$$\eta_{exergycon}= E_{reccon}/E_{supcon}$$

7) **Feed Water Pump 1(Process h-i):**

$$W_{actp1}= 100\vartheta_h(P_f-P_g)$$

$$W_{minp1}=e_i-e_h$$

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$$ED_{p1}=(\dot{m}_s-\dot{m}_1-\dot{m}_2)(W_{actp1}-W_{minp1})$$

$$\eta_{exergyp1}= W_{minp1}/W_{actp1}$$

8) **Feed Water Pump 2(Process j-k):**

$$W_{actp2} = 100\vartheta_j(p_c-p_f)$$

$$W_{minp2}=e_k-e_j$$

$$ED_{p2}=(\dot{m}_s-\dot{m}_1)(W_{actp2}-W_{minp2})$$

$$\eta_{exergyp2}= W_{minp2}/W_{actp2}$$

9) **Feed Water Pump 3(Process l-a):**

$$W_{actp3} = 100\vartheta_l(P_a-P_c)$$

$$W_{minp3}=e_a-e_l$$

$$ED_{p3}=\dot{m}_s(W_{actp3}-W_{minp3})$$

$$\eta_{exergyp3}= W_{minp3}/W_{actp3}$$

10) **Co-Generative heat exchanger(Process 7-8):**

$$E_{reche}=\dot{m}_w(e_{wout}-e_{w2})$$

$$E_{suphe}=\dot{m}_{g1}(e_7-e_8)$$

Vhere,

$$s_7-s_8= C_p\ln(T_7/T_8)$$

$$e_7-e_8= C_{pg}T_7- C_{pg}T_8-T_0(s_7-s_8)$$

$$s_{wout}-s_{w2}= C_p\ln(T_{wout}/T_{w2})$$

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$e_{wout}-e_{w2}= C_pT_{wout}- C_pT_{w2}-T_0(s_{wout}-s_{w2})$

$ED_{he}=E_{suphe}- E_{reche}$

$\eta_{exergyhe}= E_{reche}/E_{suphe}$

11) Exergy of Exit Gases:

$EX_{exitgas}=\dot{m}_{g1}C_{pg}[(T_8-T_0)-T_0\ln(T_8/T_0)]$

4.5 Overall Exergy Destruction:

$ED_{total}=ED_{com}+ED_{cc}+ED_{gt1}+ED_{gt2}+ED_{rh}+ED_{hrsg}+ED_{st1}+ED_{st2}+ED_{p1}+$

$ED_{p2}+ED_{p3}+ED_{fwh2}+ED_{fwh1}+ED_{con}+ED_{he}+EX_{exitgas}$

4.6 Overall Efficiencies:

4.6.1 Energy Efficiency:

$P_{gt}= W_{gt1}+ W_{gt2}- W_{com}$

$P_{st}= W_{st1}+ W_{st2}- W_{P1(act)}- W_{P2(act)}- W_{P3(act)}$

$Heat_{in}= (\dot{m}_f+\dot{m}_{f1})*CV_{fuel}$

i. Brayton Cycle:

Power Output 1 = P_{gt}

Efficiency (Brayton cycle) = Power Output1/ $Heat_{in}$

ii. Combined Cycle:

Power Output 2 = $P_{gt}+P_{st}$

Efficiency (combined cycle) = Power Output2/ $Heat_{in}$

iii. Combined Cycle with Co-Generation:

The efficiency of combined cycle with Co-Generation (overall energy efficiency) is also termed as "Utilization Factor".

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$\text{Total Output} = P_{gt}+P_{st}+ h_{ProcessWater}$

$\text{Utilization Factor} = \text{Total Output}/\ \text{Heat}_{in}$

4.6.2 Exergy Efficiency:

$EX_{supplied} = (\dot{m}_f+\dot{m}_{f1})*ex_{fuel}$

i. **Brayton Cycle:**

$EX_{rectotal1} = P_{gt}$

$\text{Exergy Efficiency (Brayton cycle)} = EX_{rectotal1}/EX_{supplied}$

ii. **Combined Cycle:**

$EX_{rectotal2} = P_{gt}+P_{st}$

$\text{Exergy Efficiency (combined cycle)} = EX_{rectotal2}/EX_{supplied}$

iii. **Combined Cycle with Co-Generation:**

$EX_{rectotal} = P_{gt}+P_{st}+\dot{m}_w C_{pw}[(T_{wout}-T_o)-T_o *(\ln(T_{wout}/T_o))]$

$\text{Overall Exergy Efficiency} = EX_{rectotal}/EX_{supplied}$

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CHAPTER-5

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5. RESULTS AND DISCUSSION

The following operating parameters are considered for analysis of the cycle.

5.1 OPERATING PARAMETERS:

A. Brayton Cycle:

Air inlet temperature=30°C.

Mass flow rate of air=1kg/s

Pressure ratio=1-8

Maximum temperature=1000°C-1200°C

Pinch Point temperature difference=25^{0}C

B. Rankine Cycle:

Inlet steam pressure= 60 bar

Saturation temperature at 60 bar=275.6^{0}C

Inlet steam temperature= 600^{0}C

Reheat steam pressure= 40 bar

Steam pressure at 1st extraction point= 50 bar

Steam pressure at 2nd extraction point= 30 bar

Steam condenser pressure= 0.095 bar

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5.2 PARAMETRIC ANALYSIS:

The combined cycle power plant augmented with reheating, regeneration and co-generation is considered for thermal analysis. The literature provides evidence that ideal regeneration can carnotize the Rankine cycle. Reheating is beneficial to the extent that the dryness fraction of steam at end of expansion will be within the allowable limits. Co-generation is a method of utilizing the waste heat from the cycle for a process application. This also has a positive effect on the performance. Co-generation is achieved by heating the cooling water from the condenser by the exhaust gases at the outlet of HRSG.

In the analysis, pressure ratio of the topping cycle and the turbine inlet temperature (TIT) are considered to evaluate the first and second law efficiencies, destruction of exergy due to irreversibilities, air-fuel ratio, Specific fuel consumption, power output of the gas & steam cycles, outlet temperature of process water etc.,

5.2.1Effect on Energy and Exergy Efficiencies:

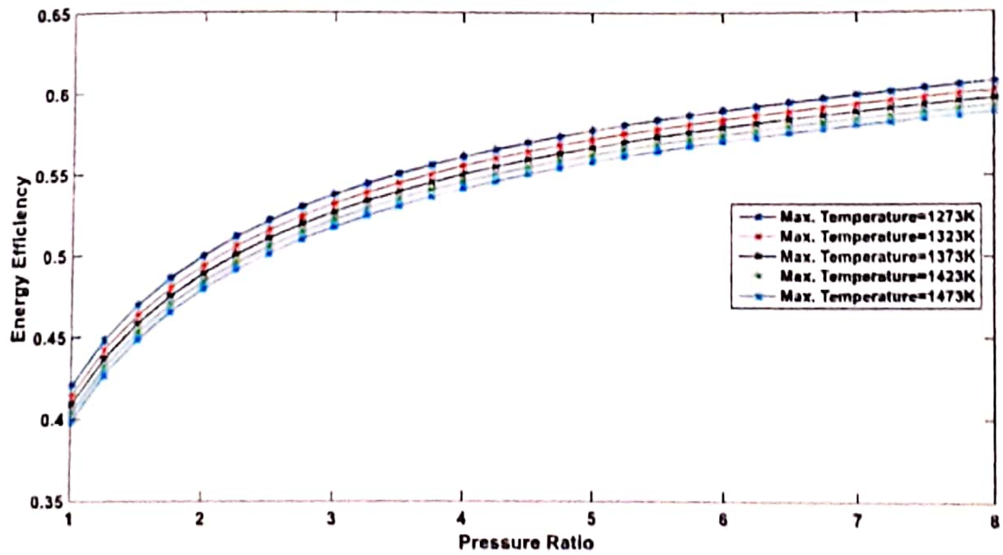

Fig 5.1 Energy Efficiency (Utilization Factor) of the complete cycle Vs. Pressure Ratio

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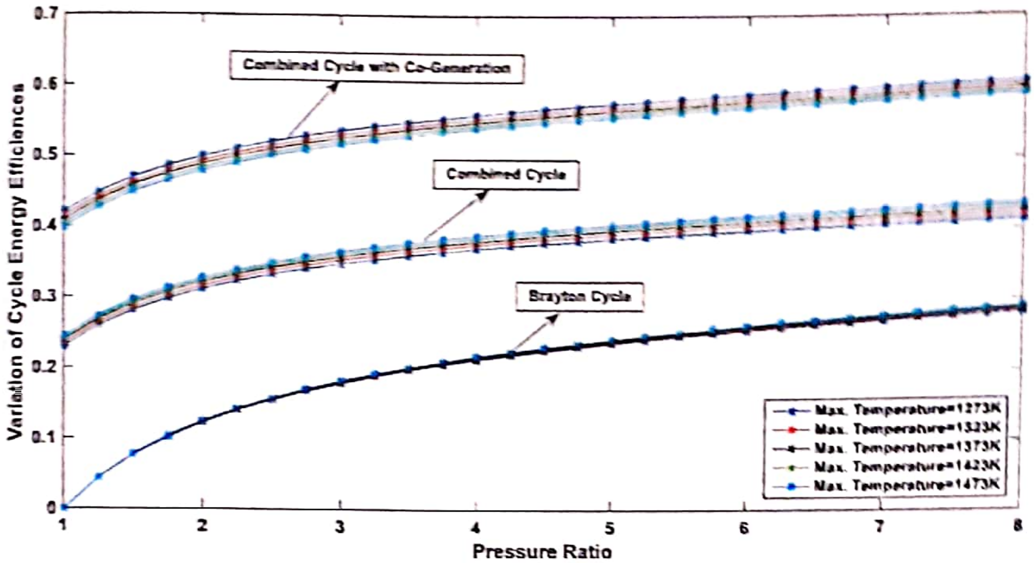

Fig 5.2Variation of cycle energy efficiencies Vs. Pressure Ratio

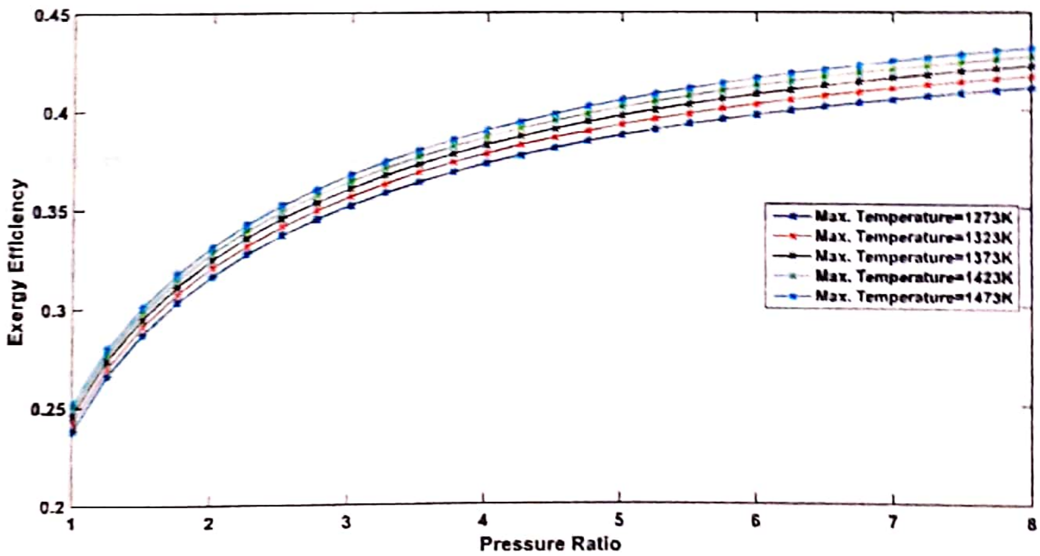

Fig 5.3Exergy efficiency of the complete cycle Vs. Pressure Ratio

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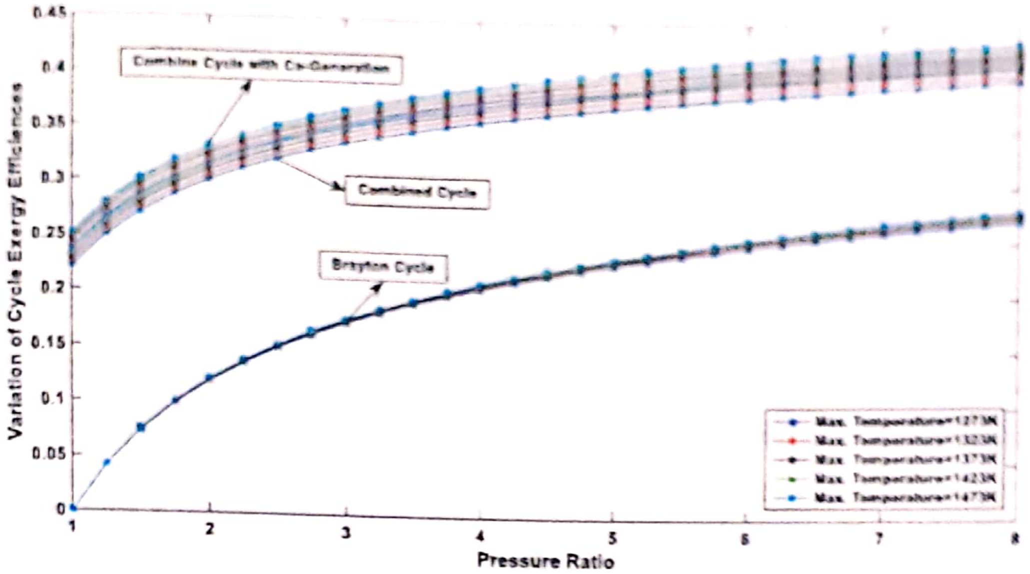

Fig 5.4Variation of cycle exergy efficiencies Vs. Pressure Ratio

The variation of energy efficiency (utilization factor) and exergy efficiency of the complete cycle with pressure ratio is plotted in figures 5.1&5.3. The curves represent the efficiencies at different turbine inlet temperatures (TIT). The figures 5.2&5.4 represent the variation of energy & exergy efficiencies considering Brayton cycle alone, combined cycle and lastly combined cycle with co-generation. There is an appreciable enhancement in energy efficiency between simple Brayton cycle & combined cycle. The maximum enhancement of 12% is observed to occur at TIT of 1473K.

The energy & exergy efficiencies include the process heat absorbed by the water. The variation of energy efficiency exhibits an interesting trend. The maximum efficiency occurs at a lower turbine inlet temperature of 1273K. This is reversed when the energy efficiency doesn't include process heat as represented in figure 5.3. In either of the cases, the efficiency is found to increase with pressure ratio. The maximum energy efficiency is 60.76% at turbine inlet temperature of 1273K by considering process heat and 43.57% without considering process heat at turbine inlet temperature of 1473K. At this temperature, the maximum energy efficiency considering process heat is only 58.89%.It is therefore observed that process heating is effective at higher turbine inlet temperatures. Considering process heat, an enhancement of 19% in energy efficiency is observed at highest turbine inlet temperature and the highest pressure ratio of 8.

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The exergy efficiency of the system reflects a similar trend like energy efficiency. The maximum exergy efficiency without process heat stands at 41.99%. Considering co-generation the efficiency stands at 43.21%. There is a marginal increase in exergy efficiencies as these figures indicate. The increase in exergy efficiency values are 14% & 2% respectively from Brayton cycle mode to combined cycle mode and from combined cycle system to the total cycle. These maximum exergy efficiencies values occur at a TIT of 1473K. From the perspective of second law, Co-generation seems to be unviable but the hot process water could be utilized for societal needs. The exergy efficiency is a direct function of maximum temperature of the cycle. Hence, it can be inferred that higher pressure ratios have to be adopted at higher temperatures.

5.2.2 Effect on Exergy Destruction:

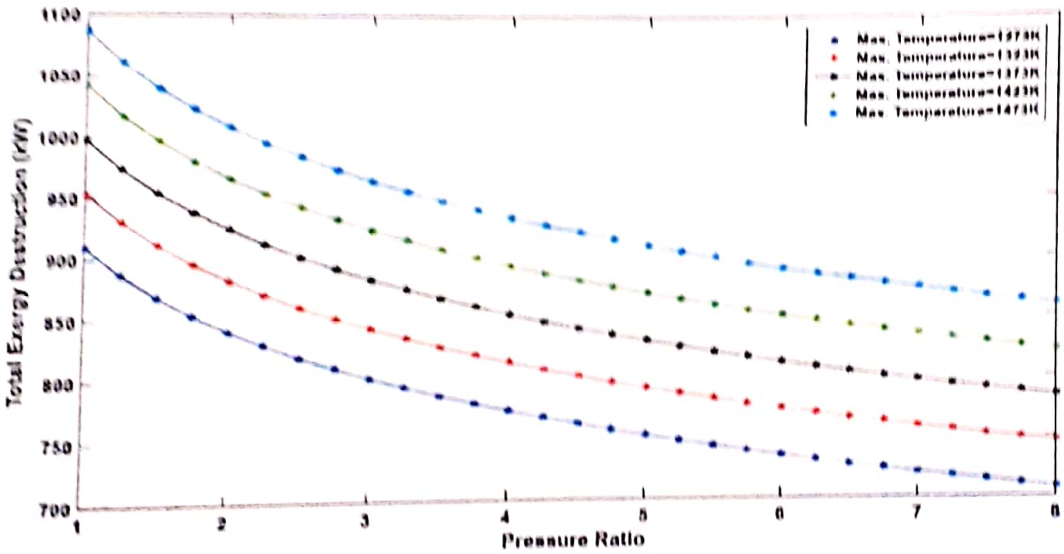

Fig 5.5Total exergy destruction Vs. Pressure Ratio

The total exergy destruction of the cycle is summation of exergy destruction of the individual components. The turbine inlet temperature and pressure ratio of the gas cycle have a major influence on the exergy destruction of the system. More than 50% of the exergy destruction is observed to occur in the main combustion chamber where the irreversible combustion process occurs.

With the increase in pressure ratio the temperature at the exit of compressor will be high. In such an eventuality, the irreversibility's associated with the combustion process is

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diminished owing to lower temperature difference between air at inlet and hot gases formed. On the contrary, an increase in TIT of the cycle will enhance the exergy losses.

5.2.3 Effect on Fuel Consumption in Combustion Chamber and Combined Re-heater:

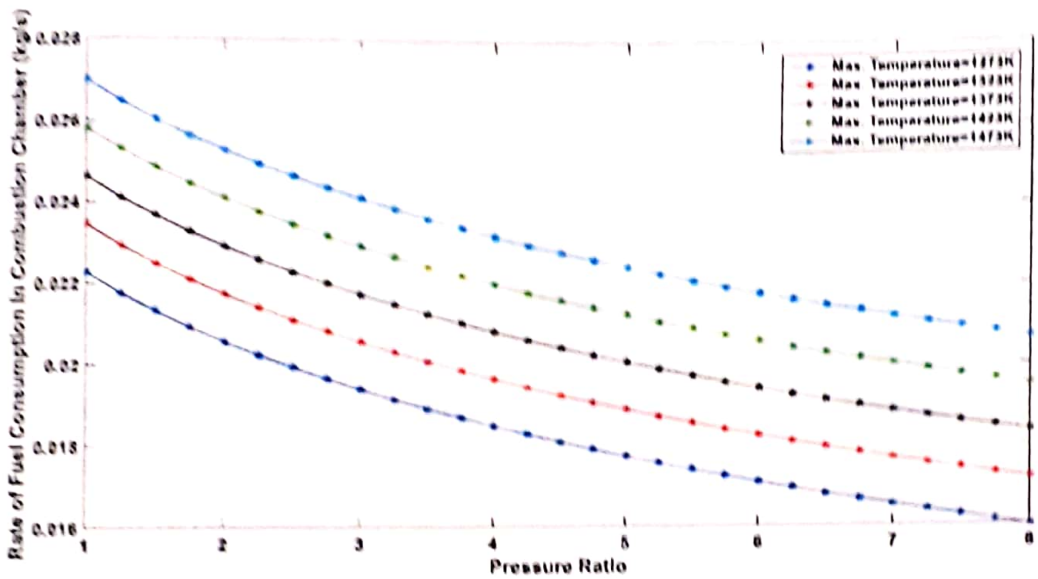

Fig 5.6 The rate of fuel consumption in main combustion chamber Vs. Pressure Ratio

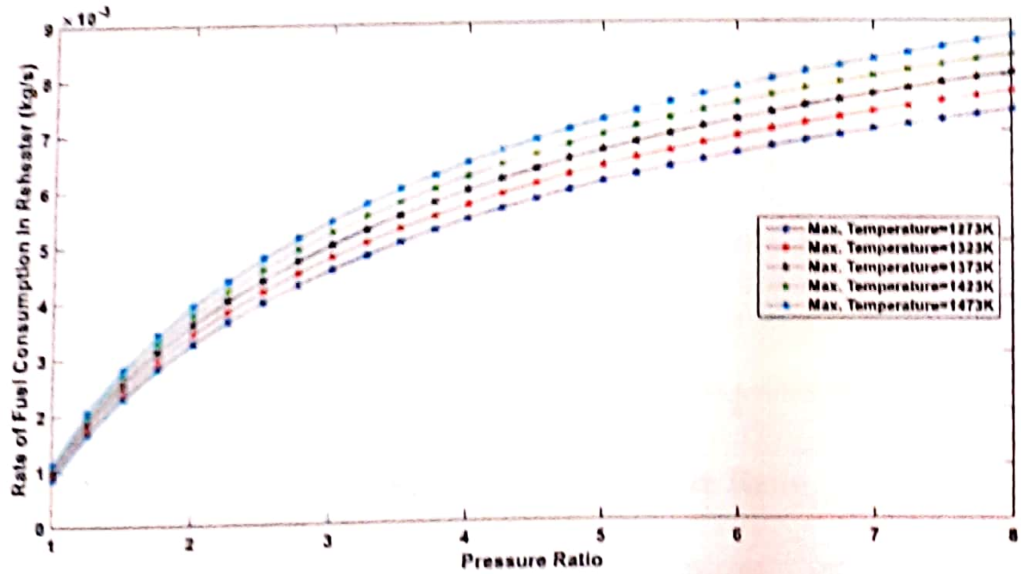

Fig 5.7 The rate of fuel consumption in Combined Reheater Vs. Pressure Ratio

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The fuel consumption in the main combustion chamber is represented in plot 5.6.The plot depicts the fact that the fuel consumption is a direct function of maximum temperature of the cycle but is inversely dependent upon the pressure ratio. With the increase in TIT the fuel consumption increases but the enhancement of pressure ratio will increase the compressor outlet temperature and thereby the fuel consumption rate decreases.

The combined cycle configuration adopted for analysis hereunder utilizes a novel method of reheating. The reheating of hot gases and steam after their first stages of expansion are carried simultaneously in the same reheater. The fuel introduced into the reheater enhances the temperature of the gases and simultaneously heats the steam indirectly to their maximum temperature. The fuel consumption in the reheater shows an increasing trend with both turbine inlet temperature and pressure ratio. This is justified as a rise in pressure ratio will reduce the temperature of gases at the end of first stage of expansion. It is also observed that the fuel consumption rate in the reheater is lesser by 1/10thof an order in comparison with the fuel consumption rate in main combustion chamber.

5.2.4. Effect on Air-Fuel Ratio:

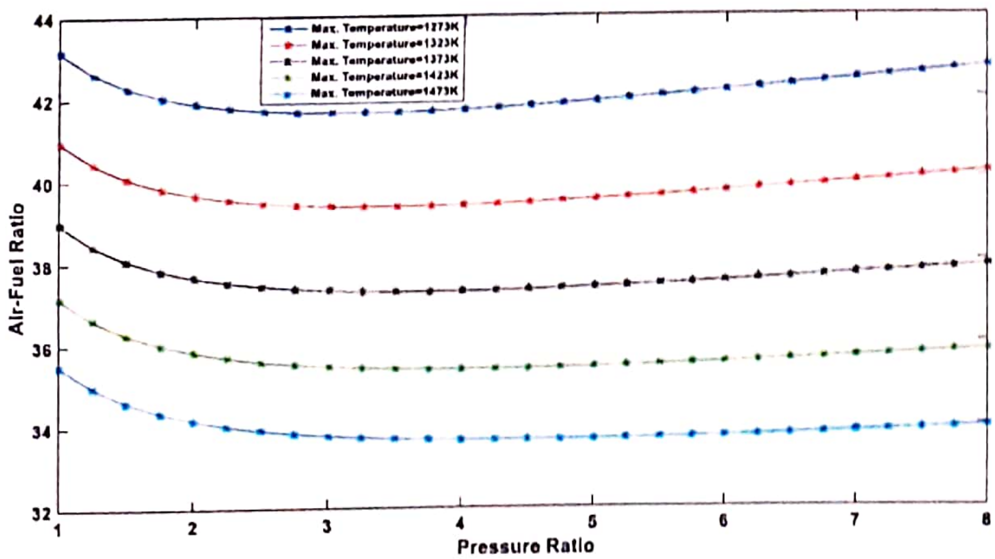

Fig 5.8 Air-Fuel ratio Vs. Pressure Ratio

The rate of consumption of fuel in the reheater increases rapidly with increase in pressure ratio initially but thereafter its stabilizes. It is also observed that the fuel consumption in

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the reheater is uninfluenced by the maximum temperature at lower pressure ratios but is significantly affected by turbine inlet temperature at high pressure ratios. The overall fuel consumption of the system shows an increasing trend at lower pressure ratios and shows a declining trend after a pressure ratio of around 3. This is exhibited through the plot of air-fuel ratio versus pressure ratio in figure 5.8. This variation can be attributed to the fact that the fuel consumption in main combustion chamber and re-heater show different trends with pressure ratio & turbine inlet temperature.

5.2.5. Effect on Co-Generation:

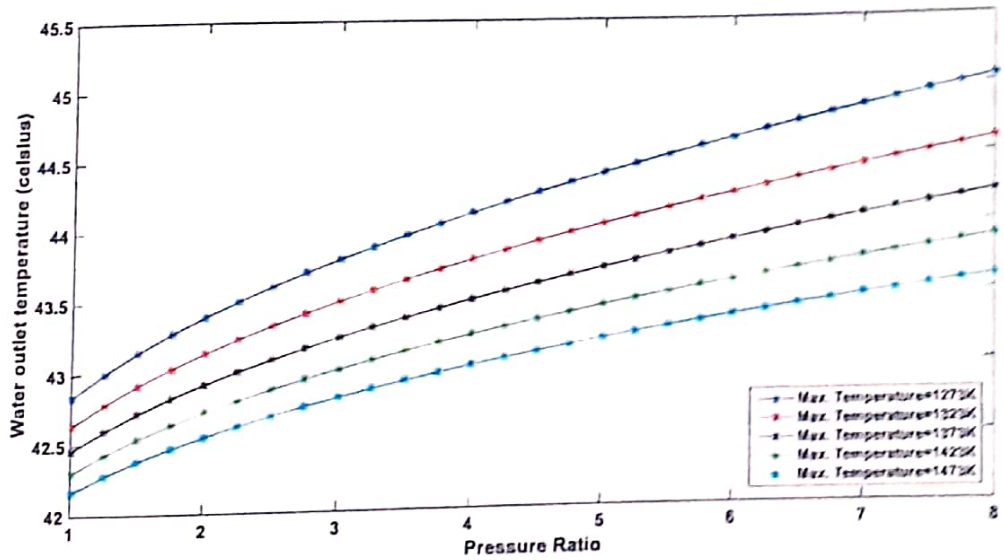

Fig 5.9 Water outlet temperature Vs. Pressure Ratio

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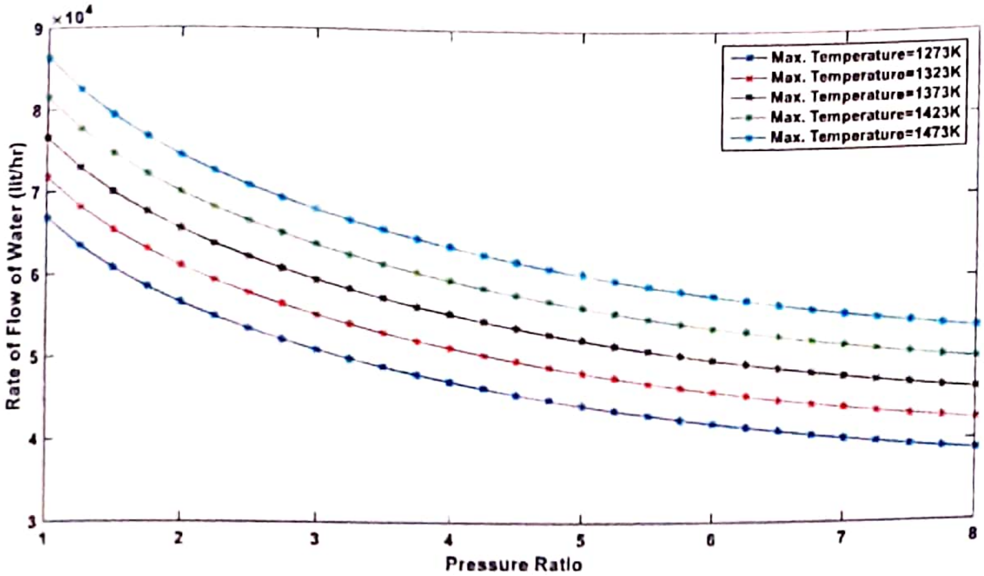

Fig 5.10 The rate of flow of process water Vs. Pressure Ratio

The combined cycle system produces power alone and the heat content of the gases at the exit of the system is generally in higher proportion. Combined cycle configuration can be augmented by Co-generation to enhance its energy utilization. This idea was implemented by introducing a heat exchanger in the line of hot gases after they have passed through the heat recovery steam generator [HRSG]. The cooling water from the condenser exits with a 5^0Crise in temperature which is further warmed up by the waste hot gases. The hot water is a by-product of the complete system encompassing Co-generation. It is beneficial to the effect that it can be used as a locality water heating system.

The results pertaining to Co-generation are illustrated in fig 5.9&5.10. The water outlet temperature ranges from 42.1^0C to 45.1^0C for the given operating conditions. The flow rate of hot water that can be supplied through this modified system varies from 40,000 lit/hr to 88,000 lit/hr which is certainly enormous in quantity and will meet the demand of hot water supply of a major locality.

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5.2.6. Effect on Specific Power Output:

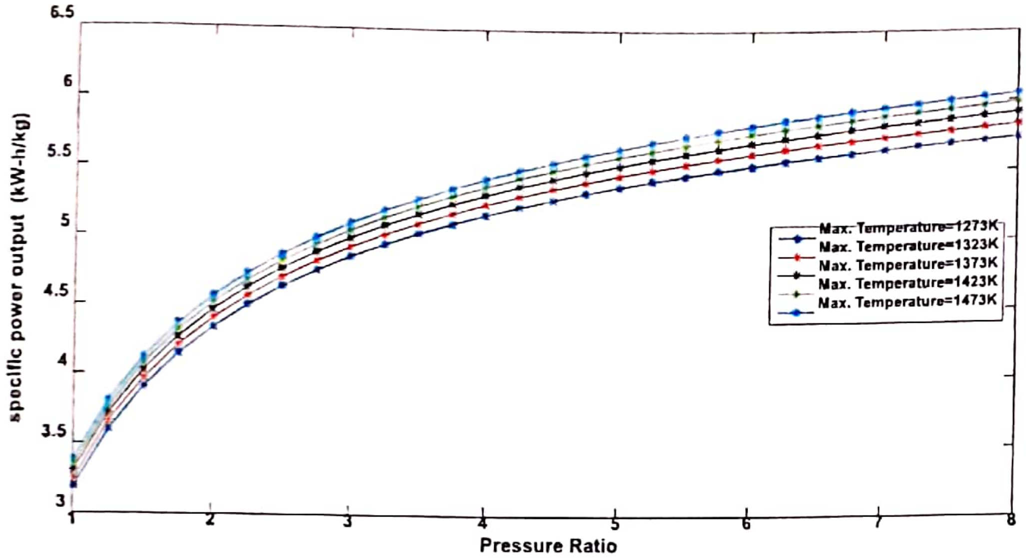

Fig 5.11.Specific power output Vs. Pressure Ratio

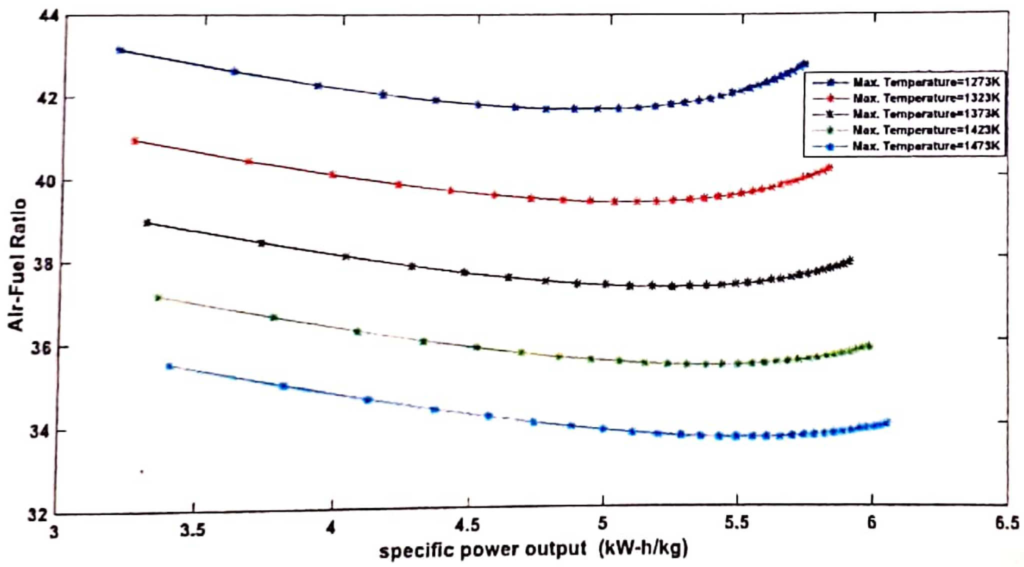

Fig 5.12.Air-Fuel ratio Vs. Specific power output

The specific power output is defined as the net power output per unit mass flow rate of fuel consumption. It's variation with pressure ratio is represented in fig5.11. It increases rapidly

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at lower pressure ratio's but thereafter exhibits a slower rate of enhancement at higher pressure ratio's. The specific power output is a direct function of turbine inlet temperature.

The variation of air-fuel ratio with specific power output (fig 5.12) exhibits an interesting trend. The air-fuel ratio at a given turbine inlet temperature decreases with increases in specific power output as it is obvious that higher fuel consumption is required at higher TIT. The air-fuel ratio reaches a minimum value at a certain specific power output indicating the highest fuel consumption. This minimum air-fuel ratio is a direct consequence of the turbine inlet temperature. The plot is indicative of the fact that the fuel consumption is maximum at a particular specific power output which itself is a function of pressure ratio. It can be inferred that the combined cycle system should operate away from this critical point. It is also necessary to operate the system towards the left of this critical point as it requires lower pressures. To the right of criticality, it can be observed that the enhancement of specific power output with the pressure ratio is very !ow and moreover higher pressure ratio entails robust construction. The pressure ratio, specific power output and air-fuel ratio at the critical point for different maximum temperatures is tabulated below.

S.No	Specific Power Output (kW-hr/kg)	Pressure Ratio	Air-Fuel ratio
1	5.03	3.5	41.66
2	5.17	3.75	39.35
3	5.31	4.0	37.27
4	5.43	4.25	35.38
5	5.60	4.75	33.66

Table 5.1 specific power output, pressure ratio and air-fuel ratio at critical points

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5.3 Apportion of Exergy Destruction in the cycle:

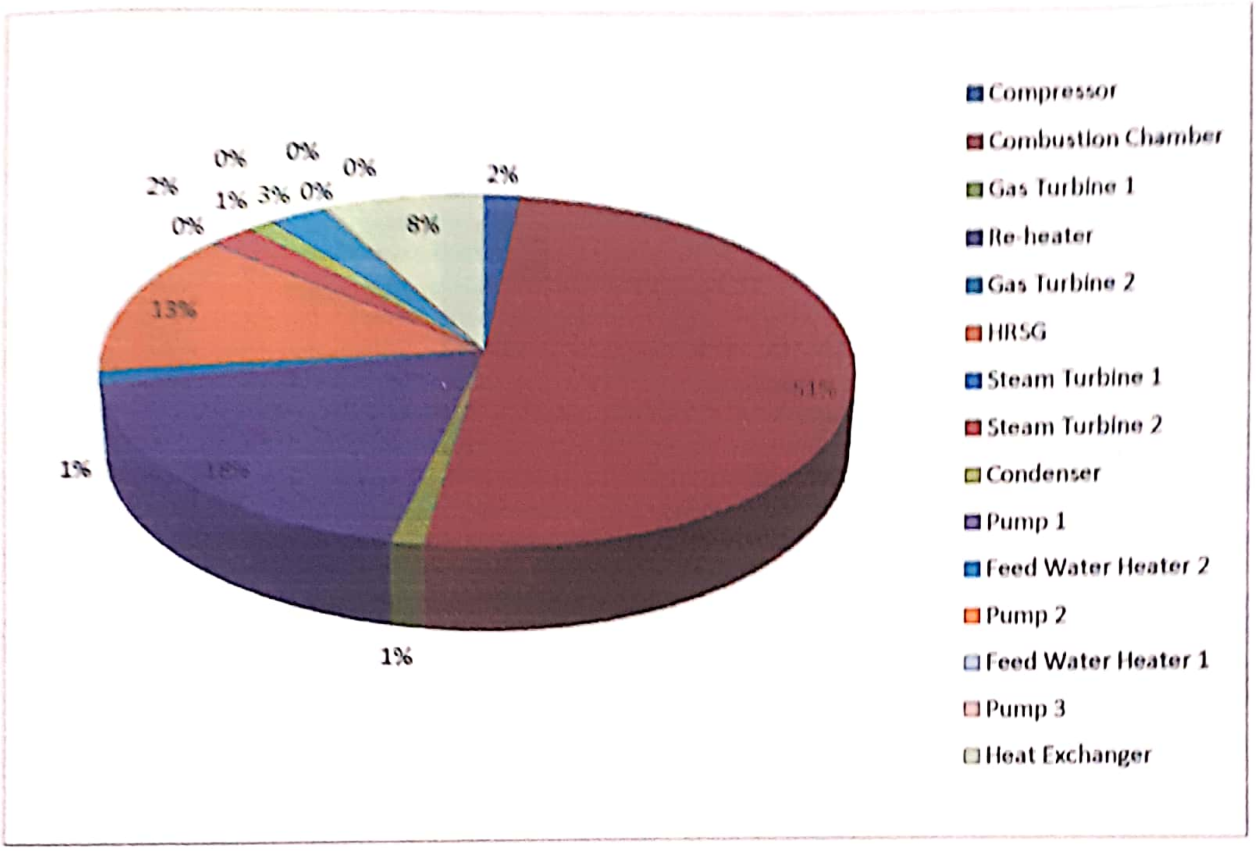

Fig 5.13 Exergy Destruction of individual components (rp=8, TIT=1473K)

The contribution of exergy destruction of individual devices is represented through a pie chart at a pressure ratio of 8 and TIT of 1473K. The major portion of exergy losses (>50%) occurs in the main combustion chamber. This inevitable exergy loss place a pivotal role in deciding the performance of the system. The second major exergy loss is in the reheater which accounts for about 18% where the reheating of working fluids is through combustion process. Hence, the combustion chambers together account for close to 70% of exergy destruction. The remaining 30% of exergy loss is mostly in the heat recovery process of HRSG and Co-generative heat exchanger which together account for 21%. Each other equipment's contribute almost evenly in the range of 0-1%.

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5.4 Exergy Efficiency of individual components:

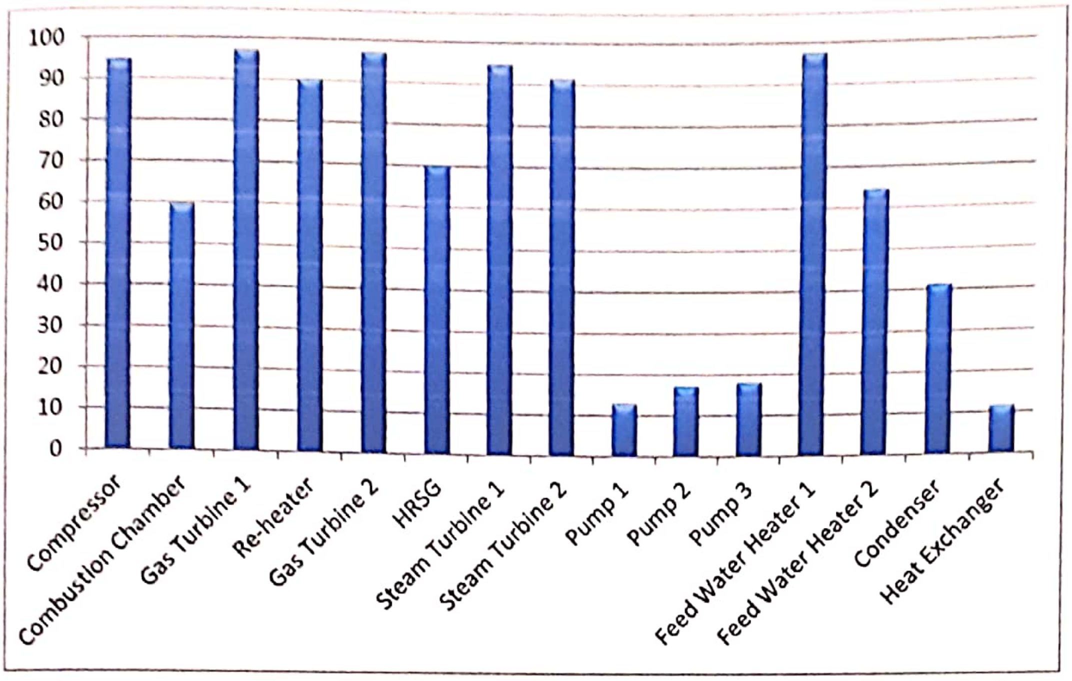

Fig 5.14 Typical Component Exergy Efficiencies (rp=8, TIT=1473K)

The typical exergy efficiency values for different equipments is depicted as a bar chart in Fig 5.15. It is observed that the pumps have the lowest exergy efficiencies in the range of 10-20%. But the exergy destruction in the pumps being of very low order, they do not influence the overall exergy efficiency of the system. The main combustion chamber is the most critical zone where it is perceived that more than 50% of total exergy loss appears. The exergy efficiency of the main combustion chamber is around 60%. The feed water heater 2 also exhibits an exergy efficiency around 65%, whereas the feed water heater 1 works with a higher second law efficiency of 97%. This abnormality in behaviour can be attributed to a wider temperature difference between the two fluid streams in FWH2. All the turbines working either on gas or steam have efficiencies above 90%.The breakup of total expansion in both the cycles into two stages could be the cause for the high efficiencies. Therefore, it is obvious that the higher the number of stages, lower will be the order of irreversibilities and increase in entropy. The condenser also accounts for a marginal exergy loss and works with an efficiency of around 40%. The Co-generative heat exchanger for warming the process water has the least second law efficiency of around 11%.

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6. CONCLUSIONS

A Combined cycle power plant system augmented with reheating, regeneration and co-generation is thermodynamically analyzed. Energy and exergy balance equations are developed for the entire system and a code has been established in MATLAB. The analysis of the cycle based on the important parameters like pressure ratio and maximum temperature was done, though there are number of other parameters affecting the cycle. The inferences drawn from the study are as given below:

- The energy and exergy efficiencies including the process heat of co-generation is observed to be higher at a lower turbine inlet temperature. This trend was reversed when process heat was not included. The energy and exergy efficiencies are also found to increase with pressure ratio.
- The variation of energy and exergy efficiencies for Brayton cycle, combined cycle and lastly combined cycle with cogeneration is also studied. The increase in energy efficiency is about 12% by switching on to combined cycle made for simple gas cycle. Further enhancement of energy efficiency is observed in transcending from combined cycle mode to combined cycle with co-generation. Similar variation is also observed in exergy efficiencies.
- The increase in exergy efficiencies considering process heat is only marginal (1.5%). Further the highest exergy efficiency occurs at the maximum turbine inlet temperature.
- The exergy destruction in the cycle is majorly contributed by the exergy loss in the combustion chambers, which is observed to be more than 50%.
- The fuel consumption rate is observed to increase with maximum temperature but is inversely proportional to the pressure ratio.
- The fuel consumption in the reheater combustion chamber is $1/10^{th}$ of the fuel consumption in the main combustion chamber.
- The uniqueness of the system is the introduction of co-generation. The hot process water is a byproduct of the system. The flow rate of hot water varies from 40,000 to 88,000lit/hr in the temperature range of 42^0C to 45^0C.

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CHAPTER-7

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7. REFERENCES

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APPENDIX

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Program for evaluation of performance of combined cycle plant

Initialization

```
exe=zeros(5,29);
ene=zeros(5,29);
ene1=zeros(5,29);
ene2=zeros(5,29);
exe1=zeros(5,29);
exe2=zeros(5,29);
rpgn=zeros(5,29);
mf1n=zeros(5,29);
mfn=zeros(5,29);
mwn=zeros(5,29);
edtotaln=zeros(5,29);
totalpowern=zeros(5,29);
ration=zeros(5,29);
ratio1n=zeros(5,29);
ratio2n=zeros(5,29);
ratio3n=zeros(5,29);
ratio4n=zeros(5,29);
tg3n=zeros(5,29);
sppoutn=zeros(5,29);
perexairn=zeros(5,29);
twoutn=zeros(5,29);
pgtn=zeros(5,29);
pstn=zeros(5,29);
afration=zeros(5,29);
```

Input properties of gas cycle

```
tg1=303;
tg8=45+273;
rp=8.0;
p1=1;
p6=p1;
tg3=1200+273;
tg5=tg3;
ma=1.0;
cpg=1.005;
r=0.287;
to=303.0;
cvf=50010.0;
exf=51888;
totalair=1;
effcom=0.9;
```

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```
effcc=0.9;
effgt1=0.9;
effgt2=0.9;
effrh=0.9;
effhrsg=0.9;
effhe=0.9;
```

Input Properties of steam cycle

```
pa=60.0;
pc=50.0;
pd=40.0;
pf=30.0;
pg=0.095;
tsb=600+273;
tsatm=275.6+273;
tgp=tsatm+25;
cpw=4.186;
tw1=35+273;
tw2=40+273;
spvh=0.001010;
spvj=0.001216;
spvl=0.001286;
effpump=0.85;
effst1=0.9;
effst2=0.9;
effcon=0.9;
```

Enthalpies

```
hsb=3656.2;
hse=3672.8;
hsfl=1213.7;
hscs=3590;
hsds=3520;
hsfs=3570;
hsgs=2340;
hsl=1154.5;
hsj=1008.3;
hsh=188.4;
```

Entropies

```
ssb=7.166;
sse=7.368;
ssc=7.175;
```

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```
        ssd=7.191;
        ssf=7.385;
        ssg=7.755;
        ssh=0.638;
        ssis=ssh;
        ssj=2.646;
        ssks=ssj;
        ssl=2.921;
        ssas=ssl;
        ssf1=3.027;

    i=1;
for tg3=1273:50:1473
    tg5=tg3;
    j=1;
for rp=1:0.25:8
    p2=rp*p1;
    p3=p2;
rpgn(i,j)=rp;
tg3n(i,j)=tg3;
```

Energy formulas

```
        phi=(0.4/1.4);
        syi=((rp^phi)-1)/effcom;
        (tg2s/tg1)=(rp)^phi
        wc=(syi*cpg*tg1);
        tg2=tg1*(1+syi);
        rho=cvf/cpg;
        rp1=rp^(0.5);
        alpha=[effgt1*(1-(1/(rp1^phi)))];
        tg6=tg3*(1-alpha);
        hsc=hsb-(effst1*(hsb-hscs));
        hsd=hsc-(effst1*(hsc-hsds));
        hsf=hse-(effst2*(hse-hsfs));
        hsg=hsf-(effst2*(hsf-hsgs));
        beta=(tg6-tgp)*[(hse-hsd)/(hsb-hsf1)]*(effhrsg);
        mf=(tg3-(tg1*(1+syi)))/((effcc*rho)-tg3);
        mf1=(1+mf)*((tg5-tg4)+beta)/((effcc*rho-tg5)-beta);
        mg=ma+mf;
        mg1=1+mf+mf1;
        wgt1=mg*(effgt1*cpg*tg3*(1-(1/rp1)^phi));
        wgt2=mg1*(effgt1*cpg*tg3*(1-(1/rp1)^phi));
        totalwt=wgt1+wgt2;
        wp1=100*spvh*(pf-pg);
        hsi=hsh+wp1;
```

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```
wp2=100*spvj*(pc-pf);
hsk=hsj+wp2;
wp3=[(100)*(spvl)*(pa-pc)];
hsa=hsl+wp3;
ms=((1+mf+mf1)*cpg*(tg6-tgp)*effhrsg)/(hsb-hsf1);
m1=(ms)*(hsl-hsk)/(hsc-hsk);
m2=(ms-m1)*(hsj-hsi)/(hsf-hsi);
wst1=(ms*(hsb-hsc))+((ms-m1)*(hsc-hsd));
wst2=((ms-m1)*(hse-hsf))+((ms-m1-m2)*(hsf-hsg));
mw=(effcon*(ms-m1-m2)*(hsg-hsh))/((tw2-tw1)*cpw);
wpact1=((ms-m1-m2)*(wp1/effpump));
wpact2=((ms-m1)*(wp2/effpump));
wpact3=(ms*(wp3/effpump));
teta=[ms/(mg1*cpg*effhrsg)];
tg7=[tgp-(teta*(hsf1-hsa))];
htw2=cpw*(tw2-273);
twout=(((effhe*mg1*1.005*(tg7-tg8))/mw)+htw2)/cpw;
hsteam=mw*cpw*(twout-tw2+273);
```

Entropy and Exergy Formulae

```
hg1=cpg*tg1;
hg2=cpg*tg2;
hg3=cpg*tg3;
hg4=cpg*tg4;
hg5=cpg*tg5;
hg6=cpg*tg6;
hg7=cpg*tg7;
hg8=cpg*tg8;
pi=(p1*p2)^0.5:
p4=pi;
sg1=(cpg*log(tg1/to))-(r*log(p1/p1));
sg2=(cpg*log(tg2/to))-(r*log(p2/p1));
sg3=(cpg*log(tg3/to))-(r*log(p3/p1));
sg4=(cpg*log(tg4/to))-(r*log(p4/p1));
sg5=(cpg*log(tg5/to))-(r*log(p5/p1));
sg6=(cpg*log(tg6/to))-(r*log(p6/p1));
sg7=cpg*log(tg7/to);
sg8=cpg*log(tg8/to);
eg4=hg4-(to*sg4);
eg5=hg5-(to*sg5);
eg7=hg7-(to*sg7);
eg8=hg8-(to*sg8);
hsis=wp1+hsh;
hsks=wp2+hsj;
hsas=wp3+hsl;
```

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```
ssi=(((hsi-hsis)/(hsj-hsis))*(ssj-ssis))+ssis;
ssk=(((hsk-hsks)/(hsl-hsks))*(ssl-ssks))+ssks;
ssa=(((hsa-hsas)/(hsf1-hsas))*(ssf1-ssas))+ssas;
esa=hsa-(to*ssa);
esb=hsb-(to*ssb);
esc=hsc-(to*ssc);
esd=hsd-(to*ssd);
ese=hse-(to*sse);
esf=hsf-(to*ssf);
esg=hsg-(to*ssg);
esh=hsh-(to*ssh);
esi=hsi-(to*ssi);
esj=hsj-(to*ssj);
esk=hsk-(to*ssk);
esl=hsl-(to*ssl);
wactcom=hg2-hg1;
wmincom=(hg2-hg1)-(to*(sg2-sg1));
edcom=ma*(wactcom-wmincom);
effexcom=wmincom/wactcom;
exsupcc=mf*exf;
exreccc=(((1+mf)*hg3)-hg2)-to*((1+mf)*sg3-sg2);
edcc=exsupcc-exreccc;
effexcc=exreccc/exsupcc;
wactgt1=hg3-hg4;
wmaxgt1=(hg3-hg4)-(to*(sg3-sg4));
edgt1=mg*(wmaxgt1-wactgt1);
effexgt1=wactgt1/wmaxgt1;
wactgt2=hg5-hg6;
wmaxgt2=(hg5-hg6)-(to*(sg5-sg6));
edgt2=mg1*(wmaxgt2-wactgt2);
effexgt2=wactgt2/wmaxgt2;
exsuprh=(mf1*exf)+(mg*eg4)+(ms*esd);
exrecrh=(mg1*eg5)+(ms*ese);
edrh=exsuprh-exrecrh;
effexrh=exrecrh/exsuprh;
exsuphrsg=mg1*((cpg*(tg6-tg7))-(to*(sg6-sg7)));
exrechrsg=ms*(esb-esa);
edhrsg=exsuphrsg-exrechrsg;
effexhrsg=exrechrsg/exsuphrsg;
wactst1=(ms*(hsb-hsc))+((ms-m1)*(hsc-hsd));
wmaxst1=(ms*(esb-esc))+((ms-m1)*(esc-esd));
edst1=wmaxst1-wactst1;
effexst1=wactst1/wmaxst1;
wactst2=((ms-m1)*(hse-hsf))+((ms-m1-m2)*(hsf-hsg));
wmaxst2=((ms-m1)*(ese-esf))+((ms-m1-m2)*(esf-esg));
edst2=wmaxst2-wactst2;
```

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```
effexst2=wactst2/wmaxst2;
wactp1=wp1/effpump;
wminp1=(ms-m1-m2)*(esi-esh);
edp1=((ms-m1-m2)*(wactp1-wminp1));
effexp1=wminp1/wactp1;
wactp2=wp2/effpump;
wminp2=(ms-m1)*(esk-esj);
edp2=((ms-m1)*(wactp2-wminp2));
effexp2=wminp2/wactp2;
wactp3=wp3/effpump;
wminp3=ms*(esa-esl);
edp3=(ms*(wactp3-wminp3));
effexp3=wminp3/wactp3;
exsupfwh2=((ms-m1-m2)*esi)+(m2*esf);
exrecfwh2=((ms-m1)*esj);
edfwh2=exsupfwh2-exrecfwh2;
effexfwh2=exrecfwh2/exsupfwh2;
exsupfwh1=((ms-m1)*esk)+(m1*esc);
exrecfwh1=ms*esl;
edfwh1=exsupfwh1-exrecfwh1;
effexfwh1=exrecfwh1/exsupfwh1;
sw2=(cpw*log(tw2/to));
sw1=sw2-(cpw*log(tw2/tw1));
hw1=cpw*tw1;
hw2=cpw*tw2;
ew1=hw1-(to*sw1);
ew2=hw2-(to*sw2);
exsupcon=(ms-m1-m2)*(esg-esh);
exreccon=mw*(ew2-ew1);
edcon=exsupcon-exreccon;
effexcon=exreccon/exsupcon;
swout=(cpw*log((twout+273)/tw2))+sw2;
hwout=cpw*(twout+273);
ewout=hwout-(to*swout);
exsuphe=mg1*(eg7-eg8);
exreche=mw*(ewout-ew2);
edhe=exsuphe-exreche;
effexhe=exreche/exsuphe;
exexitgas=mg1*((cpg*(tg8-to))-(to*cpg*log(tg8/to)));
```

Overall energy efficiency

```
heatin=(mf+mf1)*cvf;
pgt=wgt1+wgt2-wc;
pst=wst1+wst2-wpact1-wpact2-wpact3;
totalpower=pgt+pst+hsteam;
```

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```
energyeff=totalpower/heatin;
totalpower1=pgt;
energyeff1=totalpower1/heatin;
totalpower2=pgt+pst;
energyeff2=totalpower2/heatin;
```

Overall exergy efficiency

```
edtotal=edcom+edcc+edgt1+edgt2+edrh+edhrsg+edst1+edst2+edp1+edp2+edp3+edfwh+
        edfwh1+edcon+edhe+exexitgas;
exrectotal=pgt+pst+mw*cpw*((twout+273-to)-to*log((twout+273)/to));
exsupplied=(mf+mf1)*exf;
exergyeff=exrectotal/exsupplied;
 exrectotal1=pgt;
exergyeff1=exrectotal1/exsupplied;
exrectotal2=pgt+pst;
exergyeff2=exrectotal2/exsupplied;
sppout=(pst+pgt)/((mf+mf1)*3600);
actair=(mf+mf1)/0.05825;
perexair=((1-actair)/totalair)*100;
afratio=1/(mf+mf1);
ratio=pgt/pst;
ratio1=pgt/totalpower;
ratio2=pst/totalpower;
ratio3=hsteam/totalpower;
ratio4=mg/ms;
```

Plotting begins

```
ene(i,j)=energyeff;
exe(i,j)=exergyeff;
ene1(i,j)=energyeff1;
ene2(i,j)=energyeff2;
exe1(i,j)=exergyeff1;
exe2(i,j)=exergyeff2;
edtotaln(i,j)=edtotal;
mf1n(i,j)=mf1;
mfn(i,j)=mf;
mwn(i,j)=mw*3600;
ration(i,j)=ratio;
ratio1n(i,j)=ratio1;
ratio2n(i,j)=ratio2;
ratio3n(i,j)=ratio3;
ratio4n(i,j)=ratio4;
sppoutn(i,j)=sppout;
```

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```
perexairn(i,j)=perexair;
twoutn(i,j)=twout;
pgtn(i,j)=pgt;
pstn(i,j)=pst;
afration(i,j)=afratio;

   j=j+1;
end
   i=i+1;
end
```

GRAPHS

```
figure(1)
   plot(rpgn(1,:),ene(1,:),'b*-',rpgn(1,:),ene(2,:),'r*-',rpgn(1,:),ene(3,:),'k*-',rpgn(1,:),ene(4,:),'g*-
',rpgn(1,:),ene(5,:),'c*-')


figure(2)
   plot(rpgn(1,:),ene1(1,:),'b*-',rpgn(1,:),ene1(2,:),'r*-',rpgn(1,:),ene1(3,:),'k*-
',rpgn(1,:),ene1(4,:),'g*-',rpgn(1,:),ene1(5,:),'c*-',rpgn(1,:),ene2(1,:),'b*-',rpgn(1,:),ene2(2,:),'r*-
',rpgn(1,:),ene2(3,:),'k*-',rpgn(1,:),ene2(4,:),'g*-',rpgn(1,:),ene2(5,:),'c*-',rpgn(1,:),ene(1,:),'b*-
',rpgn(1,:),ene(2,:),'r*-',rpgn(1,:),ene(3,:),'k*-',rpgn(1,:),ene(4,:),'g*-',rpgn(1,:),ene(5,:),'c*-')


figure(3)
   plot(rpgn(1,:),exe(1,:),'b*-',rpgn(1,:),exe(2,:),'r*-',rpgn(1,:),exe(3,:),'k*-',rpgn(1,:),exe(4,:),'g*-
',rpgn(1,:),exe(5,:),'c*-')


figure(4)
plot(rpgn(1,:),exe1(1,:),'b*-',rpgn(1,:),exe1(2,:),'r*-',rpgn(1,:),exe1(3,:),'k*-
',rpgn(1,:),exe1(4,:),'g*-',rpgn(1,:),exe1(5,:),'c*-',rpgn(1,:),exe2(1,:),'b*-',rpgn(1,:),exe2(2,:),'r*-
',rpgn(1,:),exe2(3,:),'k*-',rpgn(1,:),exe2(4,:),'g*-',rpgn(1,:),exe2(5,:),'c*-',rpgn(1,:),exe(1,:),'b*-
',rpgn(1,:),exe(2,:),'r*-',rpgn(1,:),exe(3,:),'k*-',rpgn(1,:),exe(4,:),'g*-',rpgn(1,:),exe(5,:),'c*-')


figure(5)
   plot(rpgn(1,:),edtotaln(1,:),'b*-',rpgn(1,:),edtotaln(2,:),'r*-',rpgn(1,:),edtotaln(3,:),'k*-
',rpgn(1,:),edtotaln(4,:),'g*-',rpgn(1,:),edtotaln(5,:),'c*-')


figure(6)
```

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```
plot(rpgn(1,:),mfn(1,:),'b*-',rpgn(1,:),mfn(2,:),'r*-',rpgn(1,:),mfn(3,:),'k*-
',rpgn(1,:),mfn(4,:),'g*-',rpgn(1,:),mfn(5,:),'c*-')


figure(7)
   plot(rpgn(1,:),mf1n(1,:),'b*-',rpgn(1,:),mf1n(2,:),'r*-',rpgn(1,:),mf1n(3,:),'k*-
',rpgn(1,:),mf1n(4,:),'g*-',rpgn(1,:),mf1n(5,:),'c*-')


figure(8)
   plot(rpgn(1,:),afration(1,:),'b*-',rpgn(1,:),afration(2,:),'r*-',rpgn(1,:),afration(3,:),'k*-
',rpgn(1,:),afration(4,:),'g*-',rpgn(1,:),afration(5,:),'c*-')

figure(9)
   plot(rpgn(1,:),twoutn(1,:),'b*-',rpgn(1,:),twoutn(2,:),'r*-',rpgn(1,:),twoutn(3,:),'k*-
',rpgn(1,:),twoutn(4,:),'g*-',rpgn(1,:),twoutn(5,:),'c*-')


figure(10)
   plot(rpgn(1,:),mwn(1,:),'b*-',rpgn(1,:),mwn(2,:),'r*-',rpgn(1,:),mwn(3,:),'k*-
',rpgn(1,:),mwn(4,:),'g*-',rpgn(1,:),mwn(5,:),'c*-')


figure(11)
   plot(rpgn(1,:),sppoutn(1,:),'b*-',rpgn(1,:),sppoutn(2,:),'r*-',rpgn(1,:),sppoutn(3,:),'k*-
',rpgn(1,:),sppoutn(4,:),'g*-',rpgn(1,:),sppoutn(5,:),'c*-')


figure(12)
   plot(sppoutn(1,:),afration(1,:),'b*-',sppoutn(2,:),afration(2,:),'r*-',sppoutn(3,:),afration(3,:),'k*-
',sppoutn(4,:),afration(4,:),'g*-',sppoutn(5,:),afration(5,:),'c*-')
```

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